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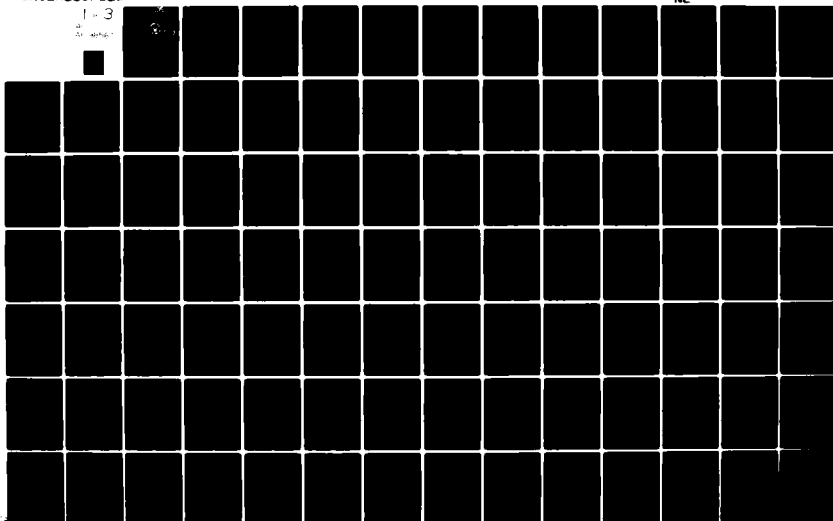
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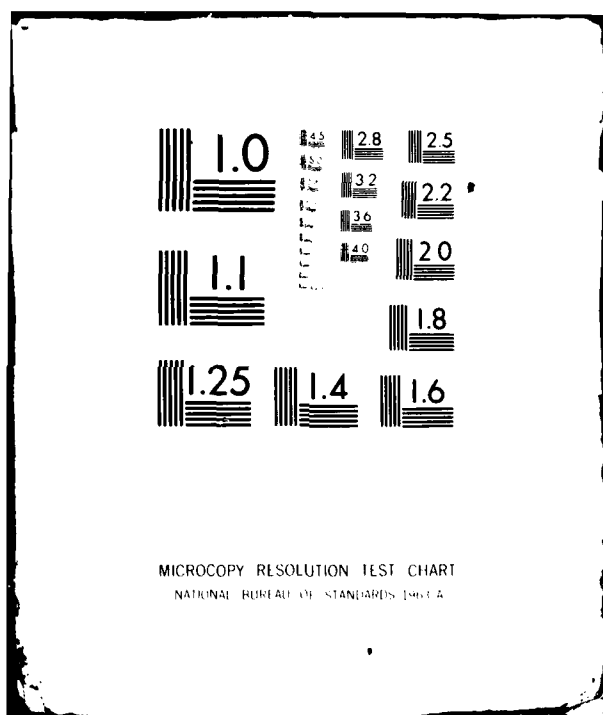
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**THESIS**

OPTIMIZATION OF A LOW  $\Delta T$  RANKINE  
POWER SYSTEM

by

Raymond C. Schaubel

December 1980

Thesis Advisor:

R. H. Nunn

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REPORT DOCUMENTATION PAGE		READ INSTRUCTIONS BEFORE COMPLETING FORM
1. REPORT NUMBER	2. GOVT ACCESSION NO.	3. RECIPIENT'S CATALOG NUMBER
	AD-A098567	(9)
4. TITLE (and Subtitle)	5. TYPE OF REPORT & PERIOD COVERED	
Optimization of a Low AT Rankine Power System.	Master's Thesis, December 1980	
6. AUTHOR(s)	7. PERFORMING ORG. REPORT NUMBER	
Raymond C. Schaubel	Delta	
8. PERFORMING ORGANIZATION NAME AND ADDRESS	9. CONTRACT OR GRANT NUMBER(s)	
Naval Postgraduate School Monterey, California 93940	(12) 236	
10. CONTROLLING OFFICE NAME AND ADDRESS	11. PROGRAM ELEMENT, PROJECT, TASK AREA & WORK UNIT NUMBERS	
Naval Postgraduate School Monterey, California 93940		
12. MONITORING AGENCY NAME & ADDRESS (if different from Controlling Office)	13. REPORT DATE	
Naval Postgraduate School Monterey, California 93940	(11) December 1980	
	14. NUMBER OF PAGES	
	235	
	15. SECURITY CLASS. (of this report)	
	Unclassified	
	16a. DECLASSIFICATION/DOWNGRADING SCHEDULE	
16. DISTRIBUTION STATEMENT (of this Report)		
Approved for public release; distribution unlimited.		
17. DISTRIBUTION STATEMENT (of the abstract entered in Block 20, if different from Report)		
18. SUPPLEMENTARY NOTES		
19. KEY WORDS (Continue on reverse side if necessary and identify by block number)		
OTEC Rankine COPES/CONMIN		
20. ABSTRACT (Continue on reverse side if necessary and identify by block number)		
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S/N 0102-014-6001

SECURITY CLASSIFICATION OF THIS PAGE (When Data Entered)

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## BLOCK 20. ABSTRACT (Continued)

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Preliminary results are presented for a range of system power levels. Optimum designs are obtained and compared for systems in which either titanium or aluminum tubes are used in the heat exchangers.

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DTIC TAB	<input type="checkbox"/>
Unannounced	<input type="checkbox"/>
Justification	
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Optimization of a Low  $\Delta T$  Rankine  
Power System

by

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Lieutenant Commander, United States Navy  
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Submitted in partial fulfillment of the  
requirements for the degree of

MASTER OF SCIENCE IN MECHANICAL ENGINEERING

from the

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## ABSTRACT

The Ocean Thermal Energy Conversion (OTEC) uses the low thermal energy potential available from ocean temperature gradients. A method is presented to analyze such systems and, for this purpose, a comprehensive simulation is developed. The simulation includes parasitic power requirements, losses due to interconnecting lines, and heat exchanger pressure drops. Cost functions are included and numerical optimization is employed to obtain optimal designs based upon minimum cost. The analysis is converted to a computer code and coupled to the COPES/CONMIN optimization code to facilitate a fully-automated design where the computer makes the design decisions and performance trade-off studies. The final product is an optimum power system module design for the designated net electrical output required and the specified system and design constraints.

Preliminary results are presented for a range of system power levels. Optimum designs are obtained and compared for systems in which either titanium or aluminum tubes are used in the heat exchangers.

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# PARTIAL LIST OF SYMBOLS

A	heat transfer surface area
$A_f$	tube bundle frontal area
$A_{ff}$	free-flow area
$C_p$	constant pressure specific heat
d	diameter
$\dot{E}$	power
f	friction factor
F	correction to LMTD
G	mass velocity
g	acceleration of gravity
$g_c$	conversion factor ( $32.2 \text{ lb}_m \cdot \text{ft} / \text{lb}_f \cdot \text{sec}^2$ )
h	specific state point enthalpy
$\bar{h}$	average heat transfer coefficient
k	thermal conductivity
$K_m$	mean salt water compressibility
L	tube or pipe length
$\dot{m}$	mass flow rate number
$N_t$	number of heat exchange tubes
$Re$	Reynolds number
P	static pressure
$\dot{Q}$	heat transfer rate
S	specific state point entropy
T	temperature
LMTD	log mean temperature difference

$U$	overall heat transfer coefficient
$v$	specific volume
$V$	velocity
$x$	quality of working fluid
$Z$	elevation
$\epsilon$	heat exchange effectiveness
$\eta$	efficiency
$\rho$	density
$\mu$	absolute or dynamic viscosity

## I. INTRODUCTION

### A. BACKGROUND

Ocean Thermal Energy Conversion (OTEC) is a concept using the low thermal energy potential available from the ocean temperature gradient that exists between warm surface ocean water and cold water in deep ocean regions.

The idea of converting the stored ocean energy to useful power originated with French physicist Jacques d'Arsonval in 1881 [Ref. 1]. It was nearly a half-century later that the technical feasibility of ocean thermal energy conversion could be demonstrated. In 1926, George Claude used an open cycle power system to extract heat from surface water for indirect conversion of the thermal energy of a working fluid. Operating at a low pressure the working fluid was used to drive a turbine providing electrical power generation.

Though Claude's limited power system produced only 22 kilowatts of electricity while requiring approximately 80 kilowatts of power to drive its equipment, it stirred the scientific and research community to consider the attractiveness of ocean thermal energy conversion [Ref. 2].

Claude called for immediate action on his ocean thermal power system, because of the Federal Oil Conservation Board's dire predictions that the United States had only six years of oil production remaining. Obviously the dire predictions ascribed to by the Federal Oil Conservation Board did not

come true, but the oil crisis of that period heightened scientific interest in extracting energy from the ocean.

Now, 55 years later, the United States is faced with an energy crisis because of increasing industrial and social dependence on foreign petroleum. Dwindling supplies and erratic price hikes have rekindled interest in ocean thermal energy conversion, since it utilizes an inexhaustible supply of fuel.

Currently, the United States Department of Energy is attempting to develop the necessary technology and demonstrate the feasibility of large-scale OTEC power systems. However, there are major engineering development problems which must be solved before OTEC can be standardized and become a viable source of electrical power generation.

The single controlling factor which creates troublesome technical encounters is low thermal power system efficiency (one to four percent depending upon parasitic power requirements). Because the heat energy used by OTEC must be extracted from a small ocean temperature difference, extremely large volumes of surface water must pass through a proportionately sized evaporator to provide sufficient indirect heat energy to convert the working fluid into vapor to drive a turbine-generator for electrical power generation. Concurrently, to convert the turbine exhaust to a saturated liquid, completing the closed cycle, a condenser having compatible heat absorption capacity must be employed.

Economic handling of the volume of fluids required for the heat absorption, expansion, and heat rejection phases of the cycle requires close scrutiny of evaporator, turbine, condenser, and pump design to minimize the parasitic losses with respect to the generated electrical output. Because of the low thermal efficiency, relative to nuclear or fossil fuel-fired power plants, the margins for design and operating error in OTEC plants will be narrow.

With the advent of high-speed computers, numerical methods for solving these complex engineering problems with multiple design variables and constraints are now possible. The case for utilizing an optimizing scheme for not just one system component, but rather the complete power generation cycle, can easily be made. In effect, it would serve as a systems analysis tool, to optimize component design and cost, relative to a specific electrical output or to enable comparison and evaluation of competing OTEC designs.

#### B. OBJECTIVES

The objectives of this work are to develop a computer code for the Ocean Thermal Energy Conversion (OTEC) power system and to couple the analysis to a numerical optimization code to provide an optimum system design capability, considering both performance and economics.

This would create an optimum modular design relative to a specified objective function for a desired net electrical output, such as a 25 MW (net) power system. Such a design



would permit construction of higher capacity power systems using the optimized modules as substations of the total power plant. Cost savings, improved plant performance, redundancy, and reliability could be the immediate beneficiaries of such a venture.

### C. OVERVIEW OF THE OTEC POWER SYSTEM ANALYSIS

To analyze the closed-cycle OTEC power system, the fundamental relationships of heat transfer, fluid mechanics and thermodynamics are used to simulate a variety of system component designs, which form the basis of the power system algorithm. The scope of this analysis will be limited to the OTEC power system and sea water systems only. Mooring systems, power delivery, hull, and cold pipe design will not be addressed.

The performance analysis will be divided into four sequential sections as shown in Figure 1, and discussed in detail in subsequent chapters of this thesis.

Input parameters (design constants) for the power cycle analysis will include:

- . Required net electrical output.
- . Salt water inlet temperature to the evaporator and condenser.
- . Length of hot and cold salt water pipes.
- . Heat exchanger tubing material (aluminum or titanium).
- . Heat exchanger tube orientation and profile.
- . Pump mechanical and motor efficiencies.

- . Turbine mechanical efficiency.
- . Generator mechanical and electrical efficiency.
- . Biofouling control factor.
- . Piping absolute roughness.
- . Projected annual inflation rate for aluminum heat exchanger retubing.

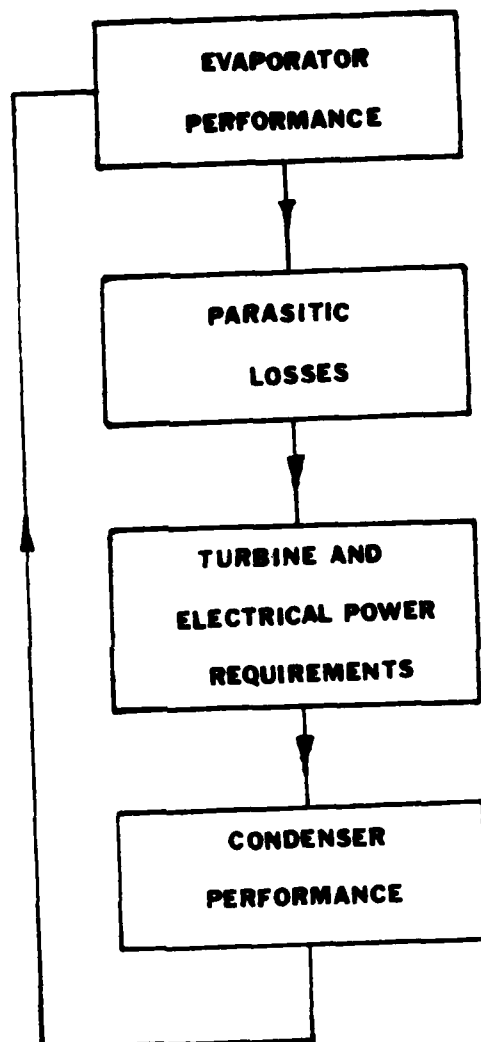


Figure 1. Power System Sequential Analysis

## II. POWER CYCLE DESCRIPTIONS

### A. INTRODUCTION

This chapter will provide a brief description of the OTEC power system. First, looking at the ideal Rankine cycle, the fundamental thermodynamic concepts will be enumerated. Then the deviations from the ideal cycle will be presented, creating the configuration assumed for the present cycle analysis which will be amplified in detail by follow-on chapters.

### B. IDEAL OTEC RANKINE CYCLE

The closed-cycle OTEC concept is based upon a Rankine power cycle that is driven by the low thermal energy potential available from the ocean temperature gradient that exists between warm surface water and cold deep water in ocean regions. The power cycle consists of a working fluid circulation pump, evaporator (heat absorption), turbine (expansion), and condenser (heat rejection), as shown in Figure 2. The majority of current OTEC designs are based upon ammonia as the working fluid -- a design decision that is adopted for this analysis.

Figure 2 also illustrates an ideal OTEC Rankine cycle, plotted on temperature-entropy coordinates. In the ideal cycle, the low pressure working fluid (state point 1) is isentropically pumped to the evaporator operating pressure (state point 2). The working fluid (ammonia) is then

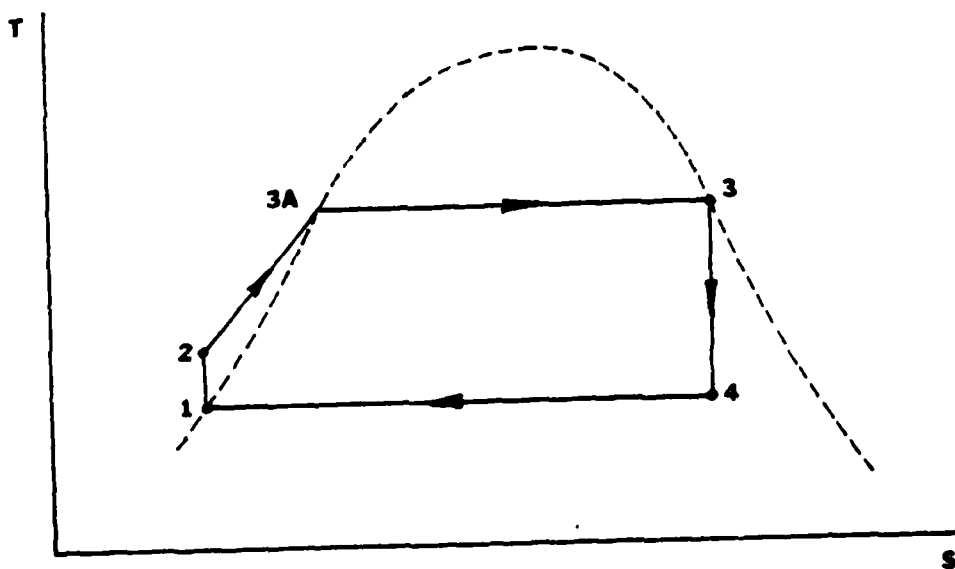
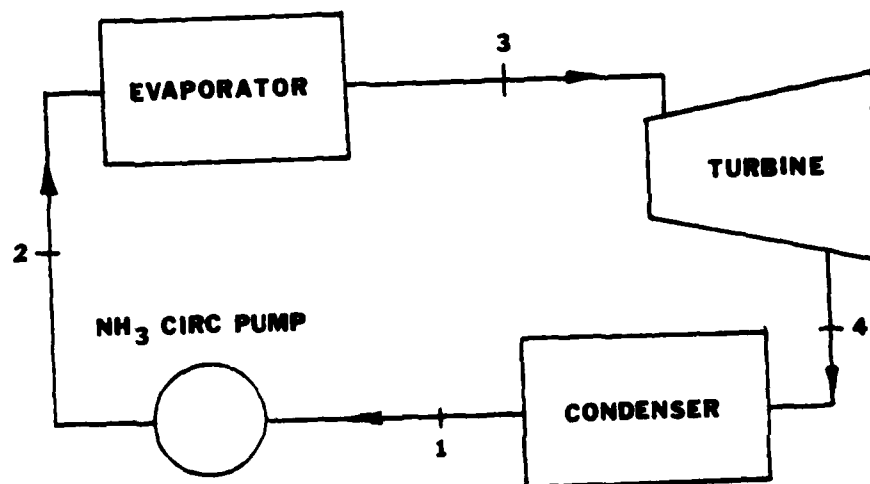


Figure 2. Idealized OTEC Rankine Cycle

converted to a saturated vapor in the evaporator by indirect heat energy exchange from warm surface ocean water (state point 3). Mechanical power is generated by isentropic expansion of the saturated ammonia vapor through the turbine (state point 4).

After exiting the turbine, the wet, low-pressure vapor is converted to a saturated liquid in the condenser by indirect heat absorption from cold ocean water (state point 1), returning the cycle back to the working fluid circulation pump.

#### C. ACTUAL OTEC RANKINE CYCLE

In actuality there are numerous deviations from the ideal cycle which must be considered in this analysis. These are:

- (1) Turbine, generator and pump efficiencies.
- (2) Pressure drops in evaporator and condenser (tube-side and shellside).
- (3) Pressure drop across moisture separator.
- (4) Elevation change and frictional losses in piping: (a) re-flux pump piping, (b) piping from circulation pump to evaporator.
- (5) Evaporator outlet quality (85 to 95%).
- (6) Moisture separator outlet quality (99 to 99.5%).

The deviations from the ideal Rankine cycle described above are depicted in the flow diagram and temperature-entropy plot of Figure 3. In the actual OTEC Rankine cycle, the low pressure working fluid (state point 1) is pumped up to the evaporator operating pressure by the ammonia circulation pump with an adiabatic efficiency (state point 2). The working

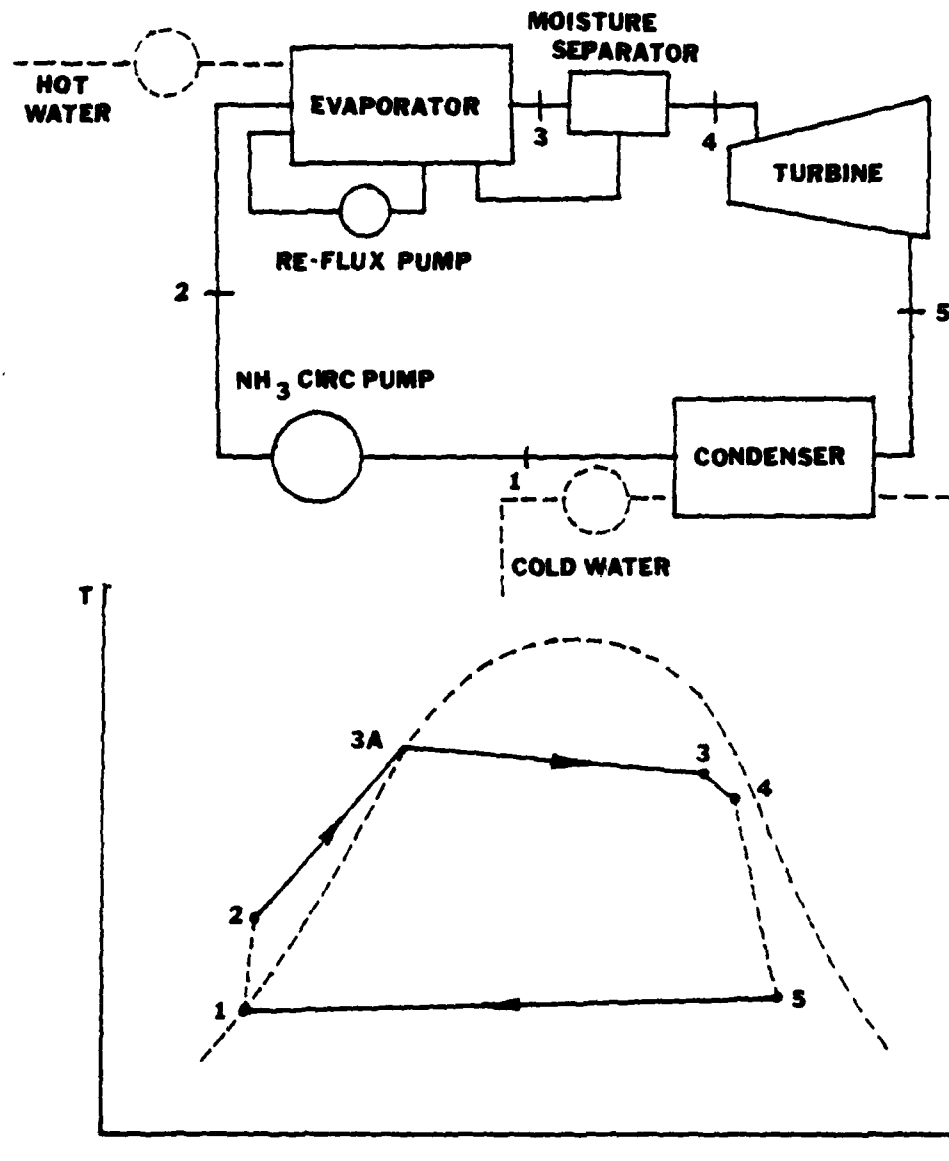


Figure 3. Actual OTEC Rankine Cycle

fluid (ammonia) is then converted to a wet vapor with an evaporator outlet quality (85-95%) acting under a shellside pressure drop (state point 3). Evaporator outlet vapor then passes through a moisture separator to improve vapor quality (99-99.5%) creating a pressure drop (state point 4). Mechanical power is generated by the expansion of the moisture separator outlet vapor through the turbine with an adiabatic efficiency (state point 5). After exiting the turbine, the wet, low pressure vapor is converted to a saturated liquid in the condenser acting under a shellside pressure drop (state point 1), returning the cycle to the working fluid circulation pump.

This figure forms the thermodynamic basis for the OTEC power system analysis which follows.



### III. EVAPORATOR AND MOISTURE SEPARATOR

#### A. INTRODUCTION

Several heat exchanger concepts have been proposed for closed-cycle OTEC systems. Among these designs are:

- . Conventional shell and tube heat exchanger.
- . Plate type heat exchanger.

Within these basic concepts, variations in design have been proposed, including:

- . Orientation of tubes (horizontal or vertical).
- . Heat exchanger tube material (i.e., titanium, aluminum).
- . Method of tube enhancement (i.e., fluted, porous coatings).
- . Location of tube enhancement (i.e., internal and/or external).
- . Location of the vapor separator (i.e., internal or external).
- . Location of the heat exchangers relative to the sea surface.
- . Method of biofouling control.

The analysis to be presented for the evaporative heat exchanger will be based on the following design characteristics:

- . Single-pass shell and tube heat exchanger.
- . Internal vapor separator with a gravity drain to evaporator inlet.
- . Horizontal orientation of tubes with an equilateral triangle or square tube profile.
- . Smooth plain-tube configuration (no enhancements).

- . Tube material (titanium or aluminum based on a 30-year life-cycle criterion).
- . Biofouling control based upon an achievable fouling factor.
- . Heat exchanger centerline located on sea surface.

As an overview of the evaporator-moisture separator analysis, the following major steps in the algorithm are listed in order of their execution (numbers in parentheses refer to equations developed in the subsequent analysis):

- . Specification of system constants (see I.C.).
- . Initialization of design variables (D.V.).
  - .. Tube length.
  - .. SW velocity through hot pipe.
  - .. Inner diameter of hot pipe.
  - .. Tube outer diameter.
  - .. SW velocity through evaporator tubes.
  - .. Inner diameter of NH3 piping.
  - .. Inner diameter of NH3 re-flux piping.
  - .. Tube profile pitch ratio.
- . Salt water mass flow rate (1).
- . Total number of tubes (2).
- . Total heat transfer surface area (3).
- . Assume an initial salt water bulk temperature (6), and ammonia heat transfer coefficient (9).
- . Overall heat transfer coefficient (4).
- . Number of transfer units (11).
- . Heat exchanger effectiveness (13).
- . Salt water outlet temperature (15).

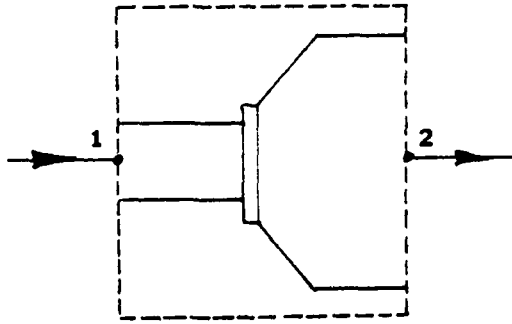
- . Revised bulk temperature (16); iterate with (6).
- . Amount of heat absorption (17).
- . Log mean temperature difference (18).
- . Film temperature (19).
- . Initial ammonia mass flow rate (21) without the effects of moisture separator.
- . Initially assume state point 1 thermodynamic properties are ideal (21).
- . Thermodynamic pump work (23).
- . Tube profile, flow parameters across the tube bank (24, etc.).
- . Tube sheet diameter (30).
- . Evaporator shellside pressure drop for two phase flow (33).
- . Moisture separator pressure drop (38).
- . Properties at state points 3 and 4 (39-41).
- . Revised ammonia mass flow rate and velocity (50) includes the effects of the moisture separator; iterate with (31).
- . Revised ammonia heat transfer coefficient (51, etc.); iterate with (9).
- . Heat exchanger cost analysis.

In the following section, the basic steps summarized above will be described in detail.

## B. ANALYSIS OF THE EVAPORATOR AND MOISTURE SEPARATOR

### 1. Salt Water Mass Flow rate, $\dot{m}_{sw}$

The salt water mass flow rate through the hot pipe must be equivalent to the flow rate through the evaporator (assuming no leakage)



$$\dot{m}_1 (\text{HOT PIPE}) = \dot{m}_2 (\text{EVAPORATOR})$$

and

$$\dot{m} = \rho_{sw} A V \quad (1)$$

where  $A$  = cross-sectional area of the hot pipe.  
 $V$  = salt water velocity through hot pipe.  
 $\rho_{sw}$  = density of salt water evaluated for an average hot pipe salt water temperature.

As previously stated, the diameter of the hot pipe and salt water velocity are among the initializing conditions of the optimization and will be treated as design variables.

## 2. Total Number of Evaporator Tubes, $N_t$

Using equation (1), it follows that

$$\dot{m}_1 = \rho_{sw} \frac{\pi d_i^2}{4} V_t N_t \quad (2)$$

where  $\rho_{sw}$  = salt water density evaluated at the average bulk temperature initially assumed as the hot pipe salt water temperature.

$d_i$  = tube inner diameter.

$N_t$  = the number of tubes required to maintain the mass flow rate for an average salt water velocity per tube.

The total number of tubes can be determined by solving Eq. (2) for  $N_t$ .

The diameter of the tube and average salt water velocity per tube are initialized for the analysis and will be treated as design variables by the optimization code.

### 3. Total Evaporator Heat Transfer Surface Area (Outer), $A_t$

Having determined the number of evaporator tubes, the total heat transfer surface area can be determined using initializing values of outer tube diameter and tube length.

For tubes without extended surfaces

$$A_t = \pi d_o L_t N_t \quad (3)$$

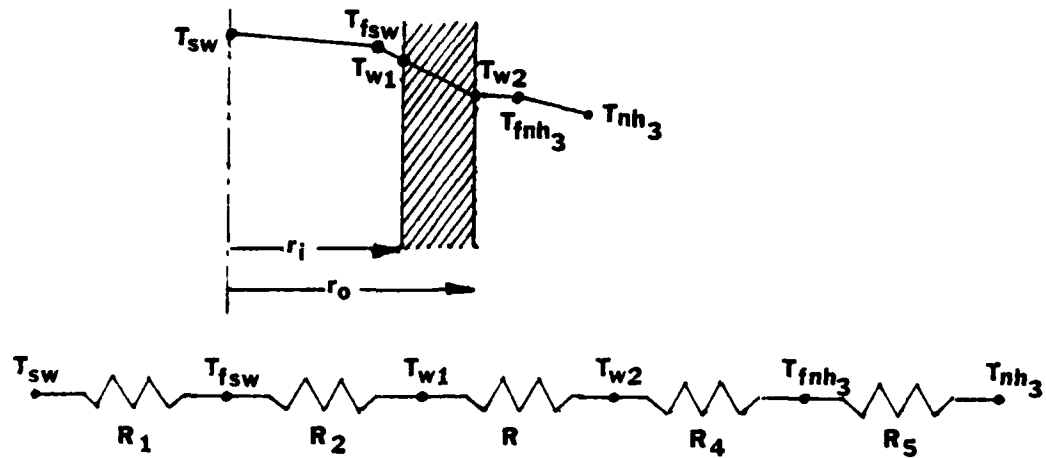
As previously, the outer tube diameter and tube length are initializing conditions and will be treated as design variables.

### 4. Overall Heat Transfer Coefficient, $U$

The quantity "U" provides a measure of the total thermal resistance in the flow path, based on either inside or outside surface area.

This analysis will be based on the value of U for the outside surface area derived from Eq. (3).

Using a resistance analysis, assuming one dimensional (radial) heat flow,



the overall heat transfer coefficient may be expressed as

$$U_o = \frac{1}{\frac{A_o}{\eta_o h_{sw} A_i} + \frac{A_o}{A_i} R_{fsw} + \frac{d_o \ln d_o / d_i}{2K} + R_{fnh_3} + \frac{1}{\eta_o h_{nh_3}}} \quad (4)$$

- where
- $h_{sw}$  = tubeside heat transfer coefficient.
  - $R_{fsw}$  = salt water fouling heat transfer resistance.
  - $K$  = thermal conductivity of the tube material.
  - $d_o, d_i$  = outer and inner tube diameter.
  - $R_{fnh_3}$  = ammonia fouling heat transfer resistance (assumed to be negligible).
  - $\eta_o, \eta_i$  = outer and inner total fin efficiency (for plain tube analysis, total fin efficiency equals 1).
  - $A_o$  = total outer surface area (including fin and bare tube).

$A_i$  = total inner surface area (including fin and bare tube).<sup>1</sup>

$$\eta_i = 1 - \frac{A_{fni}}{A_i} (1 - \eta_{fi})$$

$$\eta_o = 1 - \frac{A_{fno}}{A_o} (1 - \eta_{fo})$$

where  $A_{fni}$  = total inner fin surface area.

$A_i$  = total inner surface area (including fin and bare tube).

$A_{fno}$  = total outer fin surface area.

$A_o$  = total outer surface area (including fin and bare tube).

$\eta_{fi}$  = fin efficiency of single internal fin.

$\eta_{fo}$  = fin efficiency of single external fin.

a. Tubeside Reynolds Number,  $Re_i$

Since the heat transfer coefficient correlations for the evaporator and condenser are dependent on tubeside flow, Reynolds number must be calculated.

The tube Reynolds number is defined as

$$Re_i = \frac{\rho_{sw} v_{sw} d_i}{\mu_{sw}} \quad (5)$$

---

<sup>1</sup>Note that this analysis will hereafter consider smooth plain tube configurations only.

where  $\mu_{sw}$  = dynamic viscosity of salt water.

$\rho_{sw}$  = density of salt water.

Initially, properties are evaluated for

$$T_{BULK} = T_{sw} (INLET) \quad (6)$$

Reynolds numbers greater than 2300 will be indicative of turbulent flow [Ref. 3]. Transition flow was considered laminar for numerical evaluation.

b. Salt Water Heat Transfer Coefficient,  $h_{sw}$

The simple empirical relation proposed by Sieder and Tate [Ref. 3], expressed as

$$Nu_d = 1.86 (Re_d Pr)^{1/3} \left( \frac{d_i}{L_t} \right)^{1/3} \left( \frac{\mu}{\mu_w} \right)^{0.14} \quad (7)$$

was used for laminar heat transfer in tubes as defined by Eq. (5).

Nusselt and Prandtl numbers,  $Nu_d$  and  $Pr$ , are defined as

$$Nu_d = \frac{h_{sw} d_i}{k_{sw}}$$
$$Pr = \frac{c_{p,sw} \mu_{sw}}{k_{sw}}$$

where  $\mu_{sw}$ ,  $c_{p,sw}$  and  $k_{sw}$  (dynamic viscosity, specific heat, and thermal conductivity) of salt water are evaluated at salt water bulk temperature.



The effect of the viscosity ratio term in  
Eq. (7)

$$\left( \frac{\mu}{\mu_w} \right)^{0.14}$$

where  $\mu_w$  is salt water viscosity evaluated at tube wall temperature, is considered negligible and will hereafter be dropped from the expression of Eq. (7).

Relation (7) is based upon the following assumptions:

- . fully developed flow in smooth tubes.
- . fluid properties are evaluated at the bulk fluid temperature.

and is valid for the following condition

$$Re_d Pr \frac{d}{L} > 10$$

For fully developed turbulent flow in a tube as defined by Eq. (5), the Dittus-Boelter correlation [Ref. 3] expressed as

$$Nu_d = 0.023 Re_d^{0.8} Pr^{0.4} \quad (8)$$

was used. Nusselt and Prandtl numbers,  $Nu_d$  and  $Pr$ , are previously defined by Eq. (9).

Relation (8) is based upon the following assumptions:

- . fully developed flow in smooth tubes.
- . fluid properties are evaluated at the bulk fluid temperature

and is valid for the following conditions:

- . Prandtl numbers ranging from 0.6 to 100.
- . moderate temperature differences between the wall and fluid conditions.

c. Salt Water Fouling Heat Transfer Resistance

In this document, it will be assumed that the fouling resistance coefficient for tubeside salt water can be maintained at  $.00025 \text{ (hr.ft}^2\text{F}^\circ\text{/BTU)}$  using one of the following techniques:

- . Chlorination.
- . MAN Brush System.
- . Amertap.
- . Chemical cleaning

Pressure drops associated with cleaning techniques will not be considered in this analysis. Piping losses will be a function of tube length, inner diameter, salt water velocity and the absolute roughness of the tubing design material only.

d. Ammonia Shellside Heat Transfer Coefficient,  $h_{NH_3}$

Initially,  $h_{NH_3}$  will be assumed

$$h_{NH_3} = 1000 \text{ (BTU/hr.ft}^2\text{.F}^\circ\text{)} \quad (9)$$

since its value cannot be directly calculated during this phase of the analysis.

Using the thermal resistance expressed as

$$R_1 = \frac{d_o}{2k_i h_{sw} d_i}$$

$$R_2 = \frac{d_o}{\lambda_c h_{fsw} d_i}$$

$$R_3 = \frac{d_o \ln d_o / d_i}{2k}$$

$$R_5 = \frac{1}{\lambda_o h_{NH_3}}$$

an initial value for the overall heat transfer coefficient may be calculated.

$$U_o = \frac{1}{R_1 + R_2 + R_3 + R_5} \quad (10)$$

##### 5. NTU-effectiveness Relations

The NTU-effectiveness relationships will be used to determine the evaporator outlet salt water temperature. Currently, all salt water properties have been based upon the initial assumption that

$$T_{BULK} = T_{H_i} \text{ (SW INLET TO EVAP)}$$

The expression for the number of transfer units (NTU) which is a measure of the size of the heat exchanger is given by

$$NTU = U_o A_t / C_{min}$$

where  $C_{min}$  is defined as capacity rate of the single phase flow in an evaporative or condensing two phase flow regime.

$$C_{min} = \dot{m}_{sw} C_{p_{sw}} \quad (11)$$

Evaporator effectiveness can then be expressed as

$$\epsilon = 1 - e^{(-NTU)} \quad (12)$$

for two phase flow regardless of the flow geometry.

Using the definition of effectiveness

$$\text{Effectiveness} = \frac{\text{actual heat transfer}}{\text{maximum possible heat transfer}} \quad (13)$$

$$\epsilon = \frac{\dot{Q}}{\dot{Q}_{max}} = \frac{\Delta T_{min}}{\Delta T_{max}} = \frac{T_{Hi} - T_{Ho}}{T_{Hi} - T_{Ci}} \quad (14)$$

The expression for  $\Delta T_{min}$  represents the single phase (salt water) flow and  $T_{Ci}$  represents ammonia inlet temperature to evaporator taken at state point 3A.

#### 6. Evaporator Salt Water Outlet Temperature and Bulk Temperature

Using the relationships of Eqs. (12) and (14), the following expression may be formulated for salt water outlet temperature

$$T_{Ho} = T_{Hi} - (T_{Hi} - T_{Ci})(1 - e^{(-NTU)}) \quad (15)$$

Concurrently, a revised evaporator average salt water temperature can be expressed as

$$T_{BULK} = (T_{H_L} + T_{H_0})/2 \quad (16)$$

Using the revised value for average salt water temperature, iterate with equation (1) until the revised and current values of bulk temperature satisfy a specified convergence criterion.

7. Amount of Heat Absorption,  $\dot{Q}$

Using the results of Eq. (16) and (12), the amount of heat absorption by the evaporator may be expressed as

$$\dot{Q} = C_{min} (T_{H_L} - T_{H_0}) \quad (17)$$

8. Log Mean Temperature Difference, LMTD

The NTU-effectiveness method can be used to determine the mean effective temperature difference (LMTD) across the evaporator (heat exchanger).

Using Eq. (17) and the definition of

$$\dot{Q} = U_o A_e F LMTD$$

with  $\dot{Q}_{max} = C_{min} \Delta T_{max}$

the log mean temperature difference across the evaporator may be expressed as

$$LMTD = \frac{C_{min}(1 - e^{(-NTU)}) (T_{Hi} - T_{Ci})}{U_o A_t F} \quad (18)$$

where  $T_{Ci} = T_{NH_3}$  evaluated at state point 3.

$F$  = correction factor on  $LMTD$ , equal to 1 for two phase flow.

#### 9. Film Temperature for Property Evaluation, $T_f$

In order to evaluate the shellside ammonia heat transfer coefficient, working fluid properties (i.e., viscosity, specific heat, etc.) must be evaluated at the film temperature to validate critical heat transfer expressions.

By modifying the expression in Eq. (10) multiplying by a single tube outer area, a value for single tube conductance can be expressed as

$$U_o A = \frac{A}{R_1 + R_2 + R_3 + R_5}$$

Subsequently, the average amount of heat transferred per tube would equate to

$$\dot{Q} = U_o A (T_{BULK} - T_3)$$

where  $T_3 = T_{NH_3}$  evaluated at state point 3.

Again using the resistance analysis in Section 3, shellside wall temperature may be derived from

$$T_{W2} = T_{BULK} - \dot{Q} \left( \frac{R_1 + R_2 + R_3}{A} \right)$$

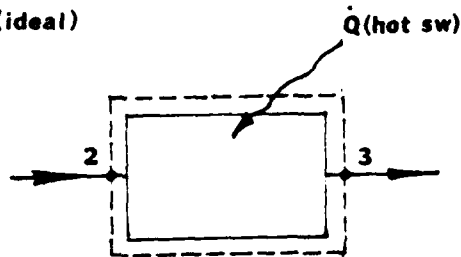
Knowing shellside wall temperature and the free-stream temperature, film temperature can be derived from their arithmetic mean.

$$T_f = \frac{T_{w_2} + T_3}{2} \quad (19)$$

10. Ammonia Mass Flow Rate,  $\dot{m}_{NH_3}$

According to first law of thermodynamics for steady state, steady-flow conditions in the evaporator:

**EVAPORATOR (ideal)**



$$\dot{m}_{NH_3} h_2 + \dot{Q} = \dot{m}_{NH_3} h_3 \quad (20)$$

from which the ammonia mass flow rate,  $\dot{m}_{NH_3}$ , may be determined if the enthalpies at state points 2 and 3 are known.

If we initialize the lower and upper bounds of the analysis in terms of pressure  $P_1$  and  $P_3$ , respectively, and initially assume that a saturated vapor leaves the evaporator, the following relations may be expressed

$$\begin{aligned} h_1 &= h_f \big|_{P_1} & T_1 &= T_{SAT} \big|_{P_1} \\ h_3 &= h_g \big|_{P_3} & T_3 &= T_{SAT} \big|_{P_3} \end{aligned} \quad (21)$$

where  $h_1$  = represents enthalpy at state point 1 at the suction inlet to the working fluid circulation pump.

$h_3$  = represents enthalpy at (ideal)/state point 3 as a saturated vapor.

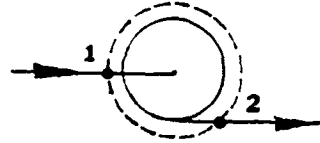
$T_1, T_3$  = represent the respective saturation temperatures.

$v_1$  = represents the specific volume at state point 1.

To summarize, the upper and lower pressure bounds of the system ( $P_1$  and  $P_3$ ) will be initialized in the analysis and treated as design variables by the optimization code. Temperature at state point 3 is initially assumed to be a saturated vapor (ideal  $T_3$ ); however, the working fluid is subject to a shellside pressure drop as it passes across the evaporator with an outlet quality of 90-95%. Properties at state point 3 (actual) will be assessed in follow-on sections.



### AMMONIA CIRC PUMP



$$\dot{m}_{NH_3} h_1 + \dot{W}_{CP} = \dot{m}_{NH_3} h_2 \quad (22)$$

Assuming steady state, steady-incompressible flow, the change in kinetic and potential energies, and heat losses are negligible for isentropic conditions, and the isentropic pump work can be expressed as

$$\dot{W}_{CPs} = v_1 (P_2 - P_1) \quad \text{where } P_2 = P_{2s}$$

After the isentropic pump work is calculated, the actual (adiabatic) pump work may be determined using pump efficiency,  $\eta_p$ .

$$\dot{W}_{CP} = \frac{\dot{W}_{CPs}}{\eta_p} \quad (23)$$

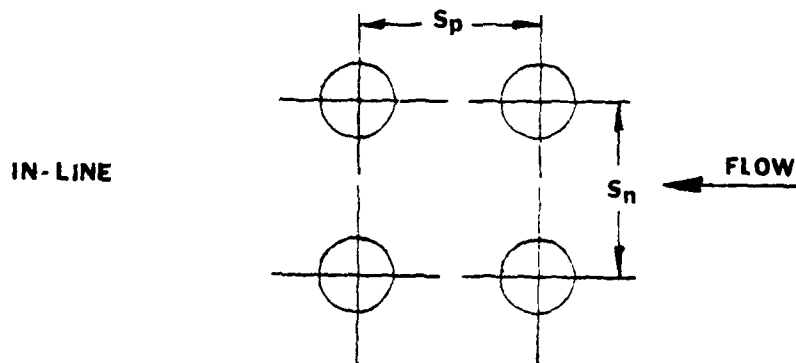
Actual outlet enthalpy at state point 2 may be determined using the results of Eq. (23) with Eq. (22) knowing the enthalpy at state point 1 from Eq. (21).

Using the results of Eqs. (21) and (22), the mass flow rate in Eq. (20) may be calculated as the average shell-side mass flow rate for the working fluid (ammonia).

# 11. Tube Profile, Flow across Tube Bank, and Tube Sheet Diameter

Since the heat-exchanger arrangements (evaporator and condenser) involve multiple rows of tubes, the geometric arrangement of the tube profiles is important in the determination of the heat transfer coefficient, the tube sheet diameter and the shell side pressure drop associated with two-phase flow (homogeneous model) [Ref. 4].

The following geometric arrangements are used:



where  $S_n$  = pitch ratio x outer tube diameter, equal to  $S_p$ .

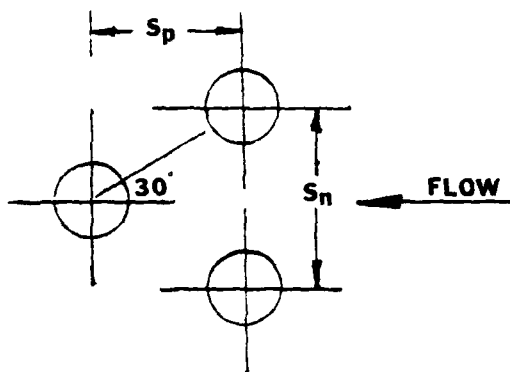
$P_R$  = pitch ratio; the distance between tube centers with respect to outer tube diameter.

$A_p$  = tube profile area (centerline to centerline) per tube.

$$S_n = P_R d_o \quad (24)$$

$$A_p = S_n^2 \quad (25)$$

STAGGERED



$$S_n = 2 P_R d_o \sin 30^\circ \quad (26)$$

$$S_p = P_R d_o \cos 30^\circ \quad (27)$$

Therefore, the tube profile area (centerline to centerline) per tube is equal to

$$A_p = S_n S_p \quad (28)$$

The ratio of minimum flow area to the frontal area can be expressed as

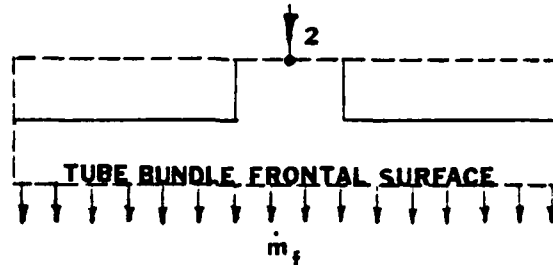
$$\frac{A_{ff}}{A_f} = \frac{S_n - d_o}{S_n} \quad (29)$$

Using the selected tube profile geometry, either in-line or staggered, and knowing the required number of tubes by equation (2), the tube sheet diameter for heat exchanger design can be assessed as follows:

$$N_t A_p = \frac{\pi T_{SD}^2}{4} \quad (30)$$

where  $T_{SD}$  = tube sheet diameter.

To estimate the shellside ammonia flow velocity the following control volume is introduced (ammonia circulation piping and the top portion of the evaporator).



If the mass flow rate remains unchanged across any boundary (continuity),

$$\dot{m}_2 = \dot{m}_f$$

Furthermore, if we assume the evaporator has the means to evenly distribute liquid droplets across the top of the tube bundle (spray nozzles and baffling), the following expressions can be applied to estimate the mean droplet velocity approaching the bundle:

Let  $(A_f)_{LiQ} = A_f \eta$

where  $\eta$  = percent of tube frontal area which is occupied by droplets.

The mass flow rates are

$$\dot{m}_2 = \rho A_p V_p$$

$$\dot{m}_f = \rho (A_f)_{L1} V_f$$

where  $A_p$  = ammonia pipe cross-sectional area.

$V_p$  = average ammonia velocity in the pipe.

Therefore

$$V_f = \frac{A_p}{(A_f) \eta_1} V_p$$

and since

$$\eta_1 \approx \frac{A_p}{A_f}$$

it follows that the average velocity of ammonia through the circulation pipe is equivalent to the average velocity of ammonia at the tube frontal area boundary.

$$V_p = V_f \quad (31)$$

Thus the assumption that  $\eta_1 = A_p/A_f$  is equivalent to the assumption of constant liquid kinetic energy in the transition from the pipe exit to the bundle entrance. Considering the minimum free-flow area for shellside flow passage,  $A_{ff}$  can be derived from Eqs. (29) and (30):

$$A_f = T_{SD} L_t$$

$$A_{ff} = A_f \left( \frac{S_{n1} - d_o}{S_n} \right) \quad (32)$$

where  $A_f$  = represents the flow frontal area.

$L_t$  = tube length.

Using the calculated values of Eqs. (32) and (20), the mass velocity for the minimum free-flow area can be expressed

$$G = \dot{m}_{NH_3} / A_{ff}$$

where  $\dot{m}_{NH_3}$  represents the average ammonia mass flow rate.

## 12. Pressure Drop of Two-Phase Flow across a Bank of Tubes, $\Delta P$

This portion of the analysis will use an analytical model for two-phase pressure drops applicable for a fog or spray flow pattern occurring at high void fractions -- the homogeneous model [Ref. 4].

The model asserts that if the pressure drop in the two-phase flow for a liquid-vapor mixture is relatively small compared to the absolute pressure, the flow is considered incompressible. Subsequently, the density of each phase is practically constant. During the process of phase change, the phase and velocity distributions are changed, and so is the momentum of the flow. Therefore, the pressure drop of a vertical two-phase flow consists of three components: friction loss, momentum change, and elevation pressure drop arising from the effects of the gravitational force field.

The local pressure gradient for a two-phase flow may be expressed as

$$\Delta P_{TOT} = \Delta P_{FRICTION} + \Delta P_{MOMENTUM} + \Delta P_{ELEVATION} \quad (33)$$

For a given channel length,  $L_c$ , the pressure drop components can be represented by

$$\Delta P_{\text{FRICTION}} = \frac{f G^2 \bar{v}}{D_e 2 g_c} L_c \quad (34)$$

$$\Delta P_{\text{MOMENTUM}} = \frac{G^2 \bar{v}}{g_c}$$

$$\Delta P_{\text{ELEVATION}} = \frac{g}{\bar{v} g_c} L_c$$

and the total pressure drop,  $\Delta P_{\text{EVAP}}$ , is given by the sum of these expressions

where  $f$  = single-phase friction factor by Jakob expressed in Eqs. (35) and (36).

$L_c$  = channel flow length, defined for horizontal tubed evaporators as  $L_c = T_{SD}$  (tube sheet diameter).

$D_e$  = equivalent diameter of flow channel, defined by  $D_e = P_R a_o - a_o$ .

$\bar{v}$  = mean specific volume defined by

$$\bar{v} = v_f \left[ 1 + \frac{\chi}{v_f} (v_g - v_f) \right]$$

where  $\chi$  = quality of mixture (state point 3).

$v_f$  = specific volume of liquid (state point 1).

$v_g$  = specific volume of vapor (state point 3).

The basic assumptions of the homogeneous model (fog flow model) [Ref. 4] are:

- (1) equal linear velocities of vapor and liquid,
- (2) thermodynamic equilibrium between the two phases, and
- (3) a suitably defined single-phase friction factor is applicable to the two-phase flow.

Using assumption (3) and the correlations by Jakob [Ref. 3], a suitable single-phase friction factor can be calculated from previously defined tube profile relationships: for staggered tube arrangements:

$$f = \left\{ 0.25 + \frac{0.118}{\left[ (S_n - d_o) / d_o \right]^{1.08}} \right\} Re_{max}^{-0.16} \quad (35)$$

and for in-line tube arrangements:

$$f = \left\{ 0.044 + \frac{0.03 S_n / d_o}{\left[ (S_n - d_o) / d_o \right]^{0.43 + 1.13 d_o / S_n}} \right\} Re_{max}^{-0.15} \quad (36)$$

where Reynolds number (max) is determined from the shellside ammonia flow and the nozzling effect of the tube geometry as expressed by

$$V_{max} = V_f \left( \frac{S_n}{S_n - d_o} \right)$$

where  $V_f$  = the ammonia velocity at the tube frontal area boundary determined by equation (31).



Reynolds number for maximum shellside flow can be calculated using the following expression

$$Re_{max} = \frac{\rho V_{max} d_o}{\mu_f} \quad (37)$$

Eq. (37) and tube profile data can then be used to evaluate the single-phase friction factor, required for Eq. (34). All other components of the total pressure drop Eq. (33) can be determined from previously calculated data.

### 13. Pressure Drop Across the Moisture Separator. $\Delta P_{m.sep}$

This portion of the analysis will simulate the use of a cyclone separator to improve the evaporator outlet vapor quality. The flow pattern in a cyclone separator is complex and simplifying assumptions are inadequate to allow the calculation of the corresponding pressure drop, which can vary from 1 to 20 inlet velocity heads [Ref. 5]. Therefore, the worst case condition will be applied with an approximation for the fluid flow inlet area to the separator banks.

By approximating the inlet area as a fraction of the evaporator frontal area

$$A_{INLET} = 0.1 T_{SD} L_t$$

the inlet fluid velocity can then be determined using the working fluid mass flow rate, Eq. (20).

$$\dot{m}_{NH_3} = \rho A_{INLET} V$$

where  $\rho$  = density of ammonia at state point 3.

Therefore, if the pressure drop across the moisture separator is equal to 20 times the inlet velocity head,

$$\Delta P_{in.sep} = 20 \rho \frac{V^2}{2g_c} \quad (38)$$

#### 14. Enthalpy at State Points 3 and 4

Since Eq. (33) represents the pressure drop across the evaporator shellside, the actual pressure at state point 3 or evaporator outlet may be determined from

$$P_{3(NEW)} = P_3 - \Delta P_{EVAP} \quad (39)$$

where  $P_3$  was previously described as the pressure for a saturated vapor.

Similarly the actual pressure at state point 4, the moisture separator outlet, may be expressed as

$$P_4 = P_{3(NEW)} - \Delta P_{in.sep} \quad (40)$$

Operating under the dome of the Temperature-Entropy diagram, the following properties are defined

$$\begin{aligned} h_{3f(NEW)} &= h_f \big|_{P_{3(NEW)}} & h_{4f} &= h_f \big|_{P_4} \\ h_{3g(NEW)} &= h_g \big|_{P_{3(NEW)}} & h_{4g} &= h_g \big|_{P_4} \end{aligned} \quad (41)$$

The subscript (NEW) representing a revised property will hereafter be dropped from the expressions in Eq. (41).

Assuming an evaporator outlet quality of 90-95%, and a moisture separator outlet quality of 99-99.5%, enthalpies at state points 3 and 4 may be determined using the relationships of Eqs. (41)

$$\begin{aligned}h_3 &= h_{3f} + x_3 (h_{3g} - h_{3f}) \\h_4 &= h_{4f} + x_4 (h_{4g} - h_{4f})\end{aligned}\tag{42}$$

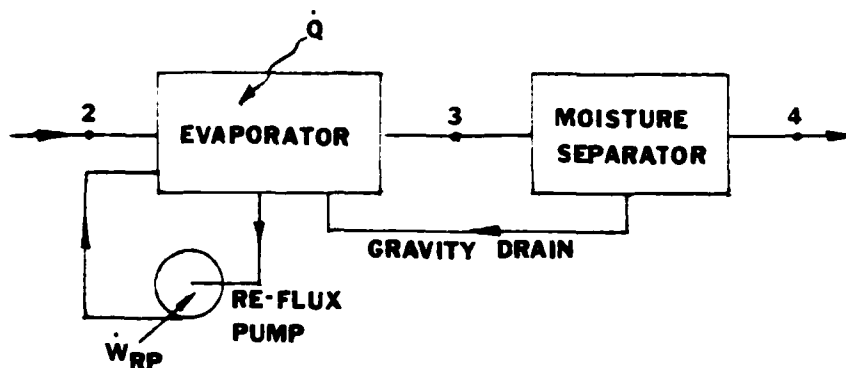
#### 15. Revised Ammonia Mass Flow Rate and Velocity

Till now, we assumed that the shellside mass flow rate was given in accordance with the ideal system defined by Eq. (20); however, in actuality this is not the case.

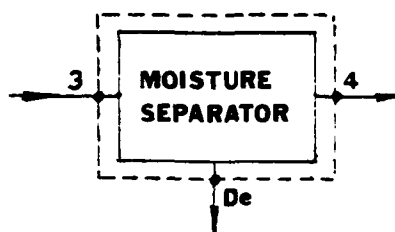
The diagrammatic representation that follows better illustrates the heat absorption phase of the OTEC power system and will provide the basis for the analysis and optimization.

Note, as in the previous control volume analysis, the following conditions are assumed.

- . Steady state.
- . Steady-incompressible flow.
- . Change in potential and kinetic energies is negligible.



Analyzing the moisture separator as a separate control volume,



If we assume that there is no carry-over of vapor in the separator drain, then

$$x_3 \dot{m}_3 = x_4 \dot{m}_4$$

and

$$\dot{m}_3 = \frac{x_4}{x_3} \dot{m}_4 \quad (43)$$

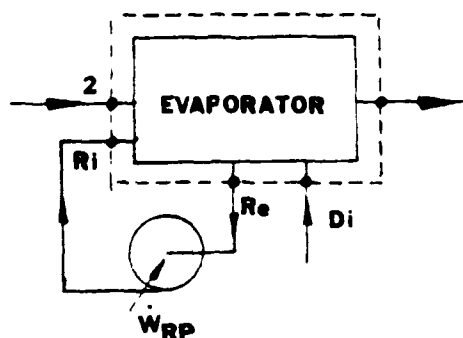
However, for reasons of flow continuity, the mass flow rate through the separator drain must be included in the control volume analysis; therefore

$$\dot{m}_3 = \dot{m}_4 + \dot{m}_{DE} \quad (44)$$

Substituting Eq. (43) into (41) and solving for  $\dot{m}_{De}$ , the following expression can be derived

$$\dot{m}_{De} = \left( \frac{K_4}{K_3} - 1 \right) \dot{m}_4 \quad (45)$$

Looking at the evaporator as a separate control volume,



the energy balance is

$$\dot{Q} + \dot{m}_2 h_2 + \dot{m}_{Re} h_{Re} + \dot{m}_{Di} h_{Di} = \dot{m}_3 h_3 + \dot{m}_{Ri} h_{Ri}$$

Assuming the change in enthalpy across the re-flux pump and the difference between the separator drain outlet and evaporator inlet are negligible, the energy balance becomes

$$\dot{Q} + \dot{m}_2 h_2 + \dot{m}_{De} h_{De} = \dot{m}_3 h_3 \quad (46)$$

where  $\dot{m}_{DC} = \dot{m}_4 \left( \frac{P_4 - P_3}{P_4} \right)$  fluid drained from the separator is

assumed to be a saturated liquid.

Furthermore, a mass balance of the evaporator control volume can be expressed as

$$\dot{m}_2 + \dot{m}_{DC} + \dot{m}_{D_1} = \dot{m}_3 + \dot{m}_{KE} \quad (47)$$

where  $\dot{m}_{KE} = \dot{m}_{D_1}$ .

Solving Eq. (47) for the mass flow rate at state point 3 and substituting into Eq. (46) with Eq. (45) yields the following expression

$$\dot{Q} + \dot{m}_2 h_2 + \left( \frac{\dot{Q}}{h_3} - 1 \right) \dot{m}_4 h_{DC} = \dot{m}_2 \frac{h_4}{h_3} \quad (48)$$

In addition, a mass balance for steady-state, steady-flow indicates that the mass flow rates at state points 2 and 4 are equal and therefore

$$\dot{m}_4 = \dot{m}_2 \quad (49)$$

Using Eqs. (48) and (49), the revised mass flow rate at state point 2 may be determined. Concurrently, the revised average ammonia velocity acting on the tube profile geometry may be determined from this revised mass flow rate.

Using the revised ammonia velocity acting on the tube profile geometry and iterating from Eq. (31) until

an acceptable convergence criterion is achieved provides the pressure drops across the evaporator and moisture separator, and the properties at state points 3 and 4 for a given film temperature. The result is more representative of the heat absorption phase in the OTEC power cycle than is the commonly used ideal analysis.

In addition, solving for the revised temperature at state point 3,

$$T_3 = T_{SAT} \bigg|_{P_3} \quad (50)$$

and iterating through Eq. (18) revises the film temperature and subsequent working fluid properties.

#### 16. Revised Shellside Ammonia Heat Transfer Coefficient

In the search for acceptable correlations to predict the average evaporative heat transfer coefficient, two analytical treatments were found that lent themselves to OTEC power system conditions.

The first of these correlations seeks to predict thin film evaporation heat transfer coefficient for horizontal tubes [Ref. 6]. Owens [Ref. 6] uses (1) the similarity between evaporation and condensation, (2) the correlation forms of local evaporation heat transfer coefficients for water on a vertical tube developed by Chun and Seban, and (3) the dependence of heat transfer on the vertical spacing of the tubes as was experimentally demonstrated by Liu, to arrive at the following correlations for non-boiling thin film evaporation:

for laminar flow

$$\bar{h} = 2.2 \left( \frac{H}{d_o} \right)^{0.1} \left( \frac{\mu_f}{g \rho_f^2 k_f^3} \right)^{-1/3} \left( \frac{4\Gamma}{\mu_f} \right)^{-1/3} \quad (51)$$

for turbulent flow

$$\bar{h} = 0.185 \left( \frac{H}{d_o} \right)^{0.1} \left( \frac{\mu_f}{g \rho_f^2 k_f^3} \right)^{-1/3} \left( \frac{C_p \mu_f}{k_f} \right)^{0.5} \quad (52)$$

where  $\frac{H}{d_o}$  = vertical spacing with respect to tube outer diameter.

$\Gamma$  = tube flow rate per unit length.

The laminar-turbulent transition point is defined by the intersection of Eqs. (51) and (52)

$$Re_{TR} = 1686 \left( \frac{C_p \mu_f}{k_f} \right)^{-1.5}$$

The pseudo-Reynolds number for horizontal vertical falling film evaporation is defined by Ref. 7.

$$Re = \frac{4\Gamma}{\mu_f}$$

The second correlation combines boiling and evaporation of liquid films on horizontal tubes, applicable for vertical banks of plain and enhanced tubes [Ref. 8].



The overall model for a single tube is expressed as

$$\bar{h} = h_b + h_d \frac{L_d}{L} + h_c \left(1 - \frac{L_d}{L}\right) \quad (53)$$

where  $h_b$  = Rohsenow pool boiling correlation over the entire tube length given by

$$h_b = \frac{\mu_f h_{fg}}{C_{sf} \sqrt[3]{\frac{J_c \sigma}{g \rho_f}}} \left( \frac{C_{pf}}{h_{fg} Pr} \right) \Delta T^2 \quad (54)$$

with  $C_{sf}$  = function of the fluid-surface combination.

$\Delta T$  = wall temperature minus free stream saturation temperature.

$\sigma_f$  = surface tension

$h_d$  = heat transfer coefficient in the developing region.

$$h_d = \frac{3}{2} C_p \frac{\Gamma}{L_d}$$

$$L_d = \frac{\Gamma^{4/3}}{4\pi \rho \alpha} \left( \frac{3 \mu_f}{g \rho_f^2} \right)^{1/3}$$

and  $h_c$  = fully developed heat transfer coefficient given for laminar flow by

$$h_c = 0.921 \left( \frac{2^2}{k^3 g} \right)^{-1/3} \left( \frac{4\Gamma}{\mu_f} \right)^{-0.22} \quad (55)$$

and, for turbulent flow,

$$h_c = 3.8 \times 10^{-3} \left( \frac{2^{-2}}{N^3 y} \right)^{-1/3} \left( \frac{4\Gamma}{\mu_f} \right)^{0.4} \left( \frac{2}{\alpha} \right)^{0.65} \quad (56)$$

where  $L$  = circumferential length of heated surface.

$\alpha$  = thermal diffusivity.

$L_d$  = developing length around tube circumference.

$\Gamma$  = flow rate per unit axial length of tube.

To apply Eq. (51) for a vertical bank of tubes,  $L$  is expressed as

$$L = N_t \pi d_o / 2$$

The laminar-turbulent transition point is defined by the intersection of Eqs. (55) and (56)

$$Re_{TK} = 5800 \left( \frac{2}{\alpha} \right)^{-1.06}$$

As before, the pseudo-Reynolds number is defined by Ref. 7

$$Re = \frac{4\Gamma}{\mu_f} \quad (57)$$

After using Eq. (57) to establish which flow regime the system is operating in, the revised heat transfer coefficient for non-boiling thin film evaporation or nucleate

boiling may be calculated and then iterated with the initial assumption for the shellside heat transfer coefficient, Eq. (9). This will have a convergence effect on variables which are a function of the shellside heat transfer coefficient, moving them closer to actual OTEC system performance characteristics.

The user should be aware that the predictions for the OTEC power system using ammonia have been for the case where no boiling occurs in the film. This condition is dictated by industrial preference for plain tube heat exchangers to minimize fouling and the characteristic of ammonia to wet surfaces well, flooding out nucleation sites. A number of enhancement techniques have been developed to create nucleate boiling, including a variety of tube configurations and surface preparations; however, a preference for them has not materialized. The nucleate boiling development in Eq. (51) which would be indicative of tube enhancement is provided for information only and will not be included in the optimization or summary of conclusions.

Having described the methods used to predict the shellside heat transfer coefficient, we can complete this chapter of the OTEC power system analysis by constructing the heat exchanger cost analysis.

#### 17. Evaporator Cost Analysis

At the request of TRW, Wyatt Industries, a large exchanger fabricator, prepared cost estimates for three different sizes of vertically configured evaporators and condensers, based upon initial design specifications prepared

by TRW. Based upon these estimates, TRW developed sets of equations that represent the costs of various heat exchanger component parts for shell diameters ranging from 10-35 ft and 35-50 ft [Ref. 9].

The following are the TRW evaporator cost (\$) equations as a function of outer tube diameter (inch), total number of tubes and tube-sheet diameter (ft) for tube-sheet diameters of 10-35 ft.

. Drilling time/tube sheet thickness

$$t_d = 0.66 (d_o - 0.5) \quad (58)$$

. Thickness of the tube sheet

$$t_{TS} = 0.56 T_{SD}^{0.68} \quad (59)$$

. Tube sheet labor cost

$$C_{TSL} = 156695 (N_t / 9630) (t_d / 0.66) (t_{TS} / 4) \quad (60)$$

. Tube sheet material cost

$$C_{TSM} = 189.486 T_{SD}^{2.3} \quad (61)$$

. Tube installation cost

$$C_{TI} = 34 N_t d_o^{0.7} \quad (62)$$

. Heat exchanger shell cost

$$C_{HXS} = 177265 \left( \frac{L_t + 6}{31} \right) (T_{SD} / 18)^2 \quad (63)$$

. Ammonia distribution plate and battles cost

$$C_{DPB} = 93865.75 (N_t / 9630) (t_d / 0.66) (T_{SD} / 18)^2 \quad (64)$$

. Bustle, flanges channels and flow plates cost

$$C_{BFCF} = 308550 (T_{SD} / 18)^2 \quad (65)$$

. Tube material cost

$$C_{TM} = (E1 L_t + E2) N_t \frac{d_o}{1.5} \quad (66)$$

where  $E1$  = curve fit of tube cost per foot.

$E2$  = tube machining cost if required

. Heat exchanger head costs

$$C_{HXH} = 53240 (T_{SD} / 18)^3 \quad (67)$$

. Water inlet, nozzles and supports cost

$$C_{WNS} = 220310.75 (T_{SD} / 18)^2 \quad (68)$$

. Tube welding costs (Titanium tubes)

for  $N_t \leq 36000$

$$C_{TW} = 14.73 N_t^{1.03} (d_o / 1.5)^{0.7} \quad (69)$$

for  $N_t > 36000$

$$C_{TW} = 0.8797 N_t^{1.3} (d_o / 1.5)^{0.7}$$

The sum of cost Eqs. (60) through (69) would equal the cost to fabricate one OTEC evaporator with a tube sheet diameter of 10-35 feet (all the preceding component costs have been adjusted for current pricing at a 10% annual rate of inflation).

If our analysis is based on a 30-year life-cycle criterion, no adjustments are necessary to any component cost equation if titanium tubing is used due to its anti-corrosive qualities; however, using aluminum tubing (i.e., Al-5052), the expense of retubing must be considered to meet the criterion of a 10-year life cycle for aluminum tubing. This implies Eq. (61) and (66) must be modified to reflect the costs of retubing at the 10 and 20-year point in the cycle.

. Aluminum tube installation cost

$$C_{ATI} = C_{TI} [1 + (1+i)^{10} + (1+i)^{20}] \quad (70)$$

where  $i$  = projected inflationary rate (input by customer)

. Aluminum tube material cost

$$C_{ATM} = C_{TM} [1 + (1+i)^{10} + (1+i)^{20}] \quad (71)$$

For tube sheet diameters of 35-50 ft the following cost relationships apply [Ref. 9]:

. Equations for drilling time/tube sheet thickness (58), thickness of tube sheet (59), and tube material costs remain unchanged.

. Tube sheet labor and material cost (titanium)

$$C_{TSL} = 55.189 N_t^{0.741} T_{SD}^{0.68} t_d \quad (72)$$

$$C_{TSM} = 29.566 T_{SD}^{2.014} t_d \quad (73)$$

. Tube sheet labor and material cost (aluminum)

$$C_{TSL} = 73.181 N_t^{0.741} T_{SD}^{0.68} t_d \quad (74)$$

$$C_{TSM} = 354.3 T_{SD}^{1.61} t_{TS} \quad (75)$$

. Tube installation costs

$$C_{TI} = 36.542 N_t d_o^{0.7} \quad (76)$$

. Heat exchanger shell cost

$$C_{HXS} = 12.544 (L_t + 6) T_{SD}^{2.06} \quad (77)$$

. Ammonia distribution plate and baffle costs

$$C_{DPB} = 158.094 T_{SD}^{1.82} + 72.419 N_t^{0.873} t_d \quad (78)$$

. Bustle, flanges, channels, and flow plate costs

$$C_{BFCF} = 472.977 T_{SD}^{2.12} \quad (79)$$

. Heat exchanger head cost

$$C_{HXH} = 1725.31 T_{SD}^{1.45} \quad (80)$$

. Water inlet, nozzles and support cost

$$C_{WINS} = 7445.297 T_{SD}^{1.1} \quad (81)$$

. Tube welding costs (titanium tubes)

for  $N_t \leq 36000$

$$C_{TW} = 14.73 N_t^{1.03} (d_o/1.5)^{0.7} \quad (82)$$

for  $N_t > 36000$

$$C_{TW} = 0.8797 N_t^{1.03} (d_o/1.5)^{0.7}$$

As indicated previously, the cost to fabricate one OTEC evaporator with a tube sheet diameter 35 to 50 ft is equal to the sum of component costs Eqs. (72) through (83) (all the preceding component costs have been adjusted for current pricing at a 10% annual rate of inflation).

For an analysis based on a 30-year system life-cycle criterion, the additional costs for aluminum retubing must be considered and Eqs. (70) and (71) apply.



#### IV. PARASITIC LOSSES

##### A. INTRODUCTION

This chapter describes in detail the programming analysis for parasitic losses which include: (1) pumping and pipe requirements for both cold and hot salt water systems, (2) pumping and pipe requirements for the working fluid (ammonia) circulation and re-flux systems, and (3) turbine generator losses due to inefficiencies. Hotel requirements have not been incorporated into the analysis, but could be included for the final design analysis.

Pumping power requirements will be determined through the use of the general energy equation between the inlet and outlet of the system control volume [Ref. 3].

$$\int_0^i \frac{dP}{\rho} + \frac{V_i^2}{2} + g Z_i = \frac{V_o^2}{2} + g Z_o + \dot{W}_s + (LOSSES)_{i \rightarrow o}$$

To determine the pumping power  $\dot{W}_s$  the following effects will be evaluated:

1. Density head.
2. Friction losses.
  - . Intake piping.
  - . Heat exchanger tubing.
  - . Exit piping (if employed).
3. Thermodynamic pressure head.
4. Elevation head.

5. Minor losses.

- . Intake piping inlet configuration (contraction).
- . Intake piping screen (obstruction).
- . Flow through valves, elbows, etc.
- . Outlet piping (expansion).
- . Inlet to heat exchanger tubing (contraction).
- . Outlet from heat exchanger tubing (expansion).
- . Outlet of exit piping (if employed).

In the above pump head evaluations, the following inputs are specified:

- . Pipe lengths (hot, cold, ammonia circulation and re-flux piping).
- . Inner pipe diameters (initialized and treated as a design variable by the optimization code).
- . Absolute roughness corresponding to piping/tubing material (designer specified).
- . Fluid velocity. (initialized and treated as a design variable by the optimization code).
- . Pump mechanical and electrical efficiencies.

As an overview of the parasitic pump loss analysis, the following major steps in the algorithm are listed in order of their execution:

- . Hot pipe salt water pump.
  - .. Inlet piping friction losses (86).
  - .. Minor piping losses due to inlet screen (87) and plenum design to evaporator core (88).
  - .. Evaporator core minor losses (89, 90) and tubeside friction losses.
  - .. Total pressure losses (92) and pumping head (93).
  - .. Pumping power requirements (95).

- .. Pump cost analysis (96).
- . Cold pipe salt water pump.
  - .. Initialize cold pipe inner diameter and SW velocity (design variables).
  - .. Minor losses due to inlet ducting (97) and plenum design to condenser core (98).
  - .. Inlet piping friction losses (99).
  - .. Condenser core minor losses (100, 101) and tubeside friction losses (103).
  - .. Density head (104).
  - .. Total pressure losses (105).
  - .. Pumping power requirement (107).
  - .. Pump cost analysis (108).
- . Ammonia circulation pump.
  - .. Piping friction (109) and minor losses due to valving/elbows (110).
  - .. Pressure drop across evaporator shellside (112).
  - .. Thermodynamic head (113).
  - .. Elevation head (114).
  - .. Total pressure losses (115).
  - .. Pumping power requirement (116).
  - .. Pump cost analysis (118).
- . Ammonia re-flux pump.
  - .. Piping friction (119) and minor losses due to valving/elbows (120).
  - .. Thermodynamic head due to pressure drop of saturated liquid ammonia across evaporator shellside (122).
  - .. Elevation head (123).
  - .. Total pressure losses (124).

- .. Pumping power requirements (126).
- .. Pump cost analysis (127).
- . Parasitic pump losses.

In the following section, the basic steps summarized above will be described in detail.

## B. ANALYSIS OF PARASITIC LOSSES

### 1. Hot Pipe Salt Water Pump, $P_{HP}$

The pressure losses due to piping friction and associated minor losses will be determined using the Darcy-Weisbach correlation [Ref. 10].

$$\Delta P = \sum_{i=1}^n \rho \frac{K_i V^2}{2g_c} \quad (83)$$

where  $K_i$  describes the resistance coefficient.

$V$  = fluid velocity.

$$K_i = f \frac{L}{D} \quad (84)$$

where  $f$  = friction factor.

$\frac{L}{D}$  = equivalent length in pipe diameters.

In order to determine the friction factor, the pipe flow Reynolds number must be calculated.

$$Re_d = \frac{\rho_{sw} V_{sw} d_i}{\mu_{sw}}$$

where  $\rho_{sw}, \mu_{sw}$  = properties of salt water at the hot pipe inlet temperature (assumed constant throughout the pipe).

$V_{sw}, d_i$  = salt water velocity and inner pipe diameter (initialized and treated as design variables by the optimization code); velocity assumed constant over pipe length.

Pipe flow Reynolds number greater than 2300 will be considered turbulent.

for laminar flow

$$f = \frac{64}{Re_d} \quad (85)$$

for turbulent flow

$$f = \frac{1.325}{\left[ \ln \left( \epsilon / 3.7 d_i + 5.74 / Re_d^{0.4} \right) \right]^2} \quad (86)$$

where  $\epsilon$  = absolute roughness corresponding to piping material selected.

Eq. (86) yields a friction factor within one percent of the Colebrook equation and is valid for the following conditions [Ref. 9].

$$10^{-6} \leq \frac{\epsilon}{D} \leq 10^{-2}$$

$$5000 \leq Re_d \leq 10^8$$

Considering the resistance coefficient for pipe minor losses

- . Assume the inlet duct is the same size as the pipe inner diameter, but it is screened

$$K = 1.5 \quad (87)$$

- . Assume piping enters evaporator through an area which is abruptly changed [Ref. 11]

$$K = \left[ 1 - \left( d_i / T_{so} \right)^2 \right]^2 \quad (88)$$

where  $T_{so}$  = evaporator tube sheet diameter (assume tube sheet diameter is twice as large as the inner pipe diameter).

Summing the results of Eqs. (84), (87), and (88) to determine the total resistance coefficient, the pressure losses due to piping can then be determined using Eq. (83).

If a variety of valves or fittings are to be included with Eq. (84), Ref. 11 provides a representative listing of equivalent length-to-pipe-diameter values.

To analyze the pressure drop across the evaporator tubeside, we again use the Darcy-Weisbach correlation, but for different design assumptions.

- . Assume inlets to evaporator tubing are well rounded [Ref. 11]

$$K = 0.5 \quad (89)$$

- . Assume outlets of evaporator tubing expand to an infinite reservoir [Ref. 10]

$$K = 1.0 \quad (90)$$

Using the Reynolds number in the previous chapter, Eq. (5), the corresponding friction factor Eq. (85) or (86), and resistance coefficient can be determined

$$K_{CORE} = f \frac{L_t}{d_i} \quad (91)$$

where  $L_t, d_i$  = evaporator tube length and inner tube diameter and are initialized and treated as design variables by the optimization code.

Summing the results of the resistance coefficient in Eqs. (89), (90) and (91), the pressure losses due to the evaporator design may be determined using the Darcy-Weisbach correlation Eq. (83).

The results of the piping losses and core design losses are equivalent to the hot pipe salt water pumping system requirements

$$\Delta P_{pump} = \Delta P_{PIPE SYSTEM} + \Delta P_{EVAP DESIGN} \quad (92)$$

converting to pumping head

$$H = \frac{g_c}{\rho_{sw} g} \Delta P_{pump} \quad (93)$$

Pumping power in terms of horsepower can be determined using the following expression

$$P_{HP} = \frac{\dot{m}_{sw}}{\eta_p} \left( \frac{g H}{g_c} \right) \quad (94)$$

where  $\eta_p$  = pump mechanical efficiency (designer input).

$\dot{m}_{sw}$  = salt water mass flow rate determined in previous chapter, Eq. (2).

To equate parasitic pump losses to power input, Eq. (94) is converted to the motor load requirement in terms of megawatts electrical.

$$P_{HP(MW)} = \frac{P_{HP}}{\eta_M} \times \text{CONVERSION FACTOR} \quad (95)$$

where  $\eta_M$  = pump motor efficiency (designer input).

Because of the high salt water flow rates and relatively low pumping heads, good engineering design would dictate the use of axial flow (propeller) type pumps.

Using the algorithm developed by TRW [Ref. 9] from data provided by Johnston Pump Co., and Process Equipment Co. (distributors of Ingersoll Rank and Johnston Pumps), the cost of salt water pumps can be expressed as

$$C_{pump} = \left[ (D/1000) 0.75 + 50 \right] 1.21 \times 10^3 \quad (96)$$



where

$$D = \frac{\pi d_i^2}{4} V_{sw}$$

where  $d_i, V_{sw}$  = inner hot pipe diameter, salt water velocity (initialized for analysis and treated as design variables by the optimization code).

The above algorithm is valid for the following conditions

- . vertical, wet pit, propeller type pumps with cast iron steel columns with protective epoxy coating, stainless steel shaft and bronze impeller.
- . pump size from 155,000 through 750,000 GPM with total dynamic heads of 8 through 12 feet.

Eq. (96) has been adjusted for current pricing at a 10% annual rate of inflation.

## 2. Cold Pipe Salt Water Pump, $P_{cp}$

Using Reynolds number

$$Re_d = \frac{\rho_{sw} V_{sw} d_i}{\mu_{sw}}$$

where  $\rho_{sw}, \mu_{sw}$  = properties of salt water at the cold pipe inlet temperature (assumed constant throughout the pipe).

$V_{sw}, d_i$  = salt water velocity and inner pipe diameter (initialized and treated as design variable by the optimization code), velocity assumed constant over pipe length.

Pipe flow characteristics and friction factor can be identified. A pumping analysis will be developed for the cold pipe pump using the Darcy-Weisbach correlation, similar to the development in the preceding section.

Considering the resistance coefficient for minor pipe losses

- . Assume the inlet duct is well rounded [Ref. 11].

$$K_{\text{INLET}} = 0.5 \quad (97)$$

- . Assume piping enters condenser through an area which is abruptly changed [Ref. 10].

$$K_{\text{PLENUM}} = \left[ 1 + (d_i / T_{SD})^2 \right]^2 \quad (98)$$

where  $T_{SD}$  = condenser tube sheet diameter (assume tube sheet diameter is twice as large as the inner pipe diameter).

- . Assume one ninety-degree elbow is required in system [Ref. 11].

$$\frac{L}{D} = 30$$

Summing the results of Eqs. (84), (97), and (98), the total resistance coefficient can be expressed as

$$K = f \left( \frac{L_P}{d_i} + \frac{L}{D} \right) + K_{\text{INLET}} + K_{\text{PLENUM}} \quad (99)$$

where  $L_p$  = length of cold pipe.

$d_i$  = inner diameter of cold pipe.

Pressure losses due to piping can then be determined using the Darcy-Weisbach, Eq. (83).

In analyzing the pressure drop across the condenser tubeside, the Darcy-Weisbach correlation is used again, but for different design assumptions.

- . Assume inlets to evaporator tubing are well rounded.

$$K = 0.5 \quad (100)$$

- . Assume outlet of condenser tubing expands to an infinite reservoir.

$$K = 1.0 \quad (101)$$

Defining Reynolds number for condenser tubeside flow, while assuming

$$T_{BULK} = T_{COLD(INLET)} \quad (102)$$

$$Re_d = \frac{\rho_{sw} V_{sw} d_i}{\mu_{sw}}$$

where  $\rho_{sw}, \mu_{sw}$  = properties evaluated at condenser tubeside bulk temperature (initially assumed equal to cold pipe inlet temperature).

$V_{sw}, d_i$  = average salt water velocity through tubing, inner condenser tube diameter (both are initialized and treated as design variables by the optimization code).

The corresponding friction factor, Eq. (85) or (86), and resistance coefficient can be determined

$$K_{CORE} = f \frac{L_t}{d_i} \quad (103)$$

where  $L_t, d_i$ , the condenser tube length and inner tube diameter are initialized and treated as design variables by the optimization code.

Summing the results of the resistance coefficient in Eqs. (100), (101), and (103), the pressure losses due to the condenser design may be determined using the Darcy-Weisbach correlation, Eq. (83).

A complete analysis of cold pipe losses must also include the effect of density head and a corresponding increase in pumping power requirements.

For most engineering problems involving the flow of liquids through a pipe, where the temperature change in the pipe is small, the density of the fluid is considered to be a constant and the fluid is termed "incompressible." However, the flow problem in OTEC cold pipe systems is unique. We can continue to assume that there is negligible change in the fluid temperature, virtually unaffected by the ocean thermal gradients, because of the system's characteristic high mass

flow rates. However, the height of the water column (1500 to 3000 feet) inside the pipe requires the effect of fluid compressibility to be taken into consideration.

The effect of an increase in density with depth can be expressed by the integral

$$\int_i^e \frac{dP}{\rho g}$$

with a density head defined as<sup>2</sup>

$$H_p = Z_e - Z_i + \int_i^e \frac{dP}{\rho g}$$

Integrating the pressure-density variation, the density head reduces to [Ref. 12]

$$H_p = Z_e - Z_i + \frac{1}{\rho_o g} (P_e - P_i) \left[ 1 - \frac{K_m}{2} (P_e + P_i) \right]$$

where  $K_m$  = mean compressibility of salt water, f(salinity, temperature and pressure).

$\rho_o$  = reference density at which  $K_m$  is evaluated.

Considering pressure at any depth obtained from the integral,

$$P = -g \int_i^e \rho(z) dz$$

the density head can be rewritten as follows

$$H_p = (Z_e - Z_i) - \frac{1}{\rho_o} \int_{Z_i}^{Z_e} \rho(z) dz \left\{ 1 - \frac{K_m}{2} \int_0^{Z_i + Z_e} \rho g dz \right\}$$

<sup>2</sup>Note that  $Z$  is measured as positive upward so that ocean depth values ( $Z_e, Z_i$ ) are negative and  $(Z_e - Z_i)$  is a positive quantity.

Rigorous procedures for calculating the density profile which is a function of temperature, salinity and pressure may be found in Ref. 13; however, they will not be discussed in this document.

For the purposes of simplification, the following solution technique was developed:

(1) If the liquid in the pipe is taken to have a constant density with respect to pressure, the compressibility approaches zero; the density head can then be expressed as

$$H_p = (z_2 - z_1) - \frac{1}{\rho_i} \int_{z_1}^{z_2} \rho(z) dz$$

(2) Converting the geometric term for elevation to an equivalent integral expression

$$z_2 - z_1 = \frac{1}{\rho_i} \int_{z_1}^{z_2} \rho_i dz$$

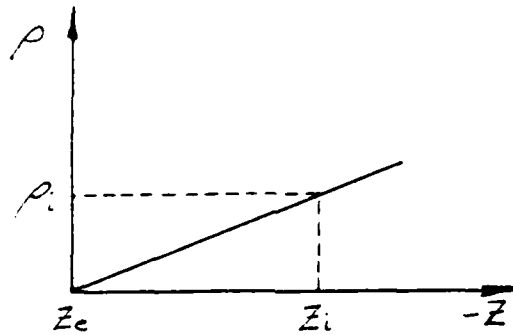
The reference density is taken to be the inlet value so that

$$\rho_i = \rho_o$$

and the density head can be rewritten as follows

$$H_p = \frac{1}{\rho_i} \int_{z_1}^{z_2} (\rho_i - \rho(z)) dz$$

(3) Assuming a linear distribution of density with depth, due to temperature variations, as illustrated below



the following linear expression for density with respect to depth may be formulated, where  $Z_e = 0$  for convenience.

$$\rho_i - \rho = (\rho_i - \rho_e)(1 - Z/Z_i)$$

(4) Applying the equation developed in section 3 to the density head integral above and integrating over the range of values for sea water depth ( $z$ ), the following equation is derived as a linear approximation to the density variation of sea water with respect to depth

$$H_p = \left( \frac{\rho_i - \rho_e}{\rho_i} \right) \left( -\frac{Z_i}{2} \right)$$

where  $\rho_i, \rho_e$  = curve fit evaluations of density for specified depths of sea water. Data extracted from Ref. 14.

The results of the piping losses, core design losses, and density head are equivalent to the cold pipe salt water pumping system requirements

$$\Delta P_{\text{pump}} = \Delta P_{\text{PIPE SYSTEM}} + \Delta P_{\text{COND DESIGN}} + \Delta P_{\text{DENSITY}} \quad (105)$$

Using Eq. (93), Eq. (105) can be converted to a pumping head. Similarly, pumping power in terms of horsepower can be determined using Eq. (44).

$$P_{CP} = \frac{\dot{m}_{SW}}{\eta_P} \left( \frac{gH}{g_c} \right)$$

where

$$\dot{m}_{SW} = \rho_{SW} \left( \frac{\pi d_i}{4} \right)^2 V_{SW} \quad (106)$$

and  $\rho_{SW}$  = density of salt water evaluated for a constant inlet temperature.

$V_{SW, di}$  = cold pipe salt water velocity, and inner diameter (initialized and treated as design variables by the optimization code). Note salt water velocity through cold pipe is considered to be constant.

Pumping power can then be expressed in terms of megawatts electrical

$$P_{CP(MW)} = \frac{P_{CP}}{\eta_M} \times \text{CONVERSION FACTOR} \quad (107)$$

where  $\eta_M$  = pump motor efficiency (designer input).

Using the same arguments for the selection of an axial flow (impeller type) pump, as used for the hot pipe salt water pump, the pump cost algorithm developed by TRW can be applied to the cold pipe salt water pump assuming



the required conditions are validated.

$$C_{\text{pump}} = \left[ (D/1000) 0.75 + 50 \right] 1.21 \times 10^3 \quad (108)$$

Equation (108) has been adjusted for current pricing at a 10% annual rate of inflation.

### 3. Ammonia Circulation Pump. $P_{\text{CIRC}}$

The function of the ammonia circulation pump is to circulate and lift saturated liquid ammonia from the condenser hot well at state point 1 and increase its pressure to exceed the operating conditions in the evaporator at state point 2.

In order to evaluate these characteristics, the following pumping elements will be included in the analysis:

- . Piping losses (friction and minor).
- . Heat exchanger shellside pressure drop.
- . Thermodynamic pressure head.
- . Elevation head.

As in the preceding analysis, Reynolds number is used to determine pipe flow characteristics

$$Re_d = \frac{\rho V d_i}{\mu}$$

where  $\rho, \mu$  = saturated liquid properties of ammonia for the temperature at state point 1 (assume any temperature increase from pump work is negligible).

$d_i$  = inner pipe diameter (initialized and treated as a design variable by the optimization code).

$V$  = ammonia flow velocity determined from the preceding chapter, Eq. (50).

The ammonia pipe friction factor can then be determined from Eqs. (85) or (86), and the piping friction resistance coefficient can be expressed as

$$K = f \frac{L}{d_i} \quad (109)$$

where  $L$  = ammonia circulation pipe length (designer input).

Considering the resistance coefficient for minor pipe losses, assume there are four ninety-degree elbows in the system

$$K = 4 \frac{L}{D} \quad (110)$$

where  $\frac{L}{D}$  = equivalent length in pipe diameters for a standard elbow [Ref. 11].

Summing the results of Eqs. (109) and (110), piping losses (friction and minor) can be determined using the Darcy-Weisbach equation (83).

$$\Delta P_{p,pe} = \rho \left[ f \frac{L}{d_i} + 4 \frac{L}{D} \right] \frac{V^2}{2g_c} \quad (111)$$

The heat exchanger shellside pressure drop is also included in the pumping head requirement because it serves as a resistance to flow.

Pressure drop across the evaporator shellside was determined using the two-phase flow model (homogeneous) expressed by Eq. (33)

$$\Delta P_{EVAP} = \Delta P_{FRICTION} + \Delta P_{MOMENTUM} + \Delta P_{ELEVATION} \quad (112)$$

Since the pump is required to lift the working fluid to a higher elevation and increase its operating pressure, the following elements must be included in the analysis:

. Thermodynamic head

$$\text{where } \Delta P_{THERMO} = P_2 - P_1 \quad (113)$$

represents the difference in thermodynamic operating pressure between state point 2 and state point 1.

. Elevation head

$$\text{where } \Delta P_{ELEVATION} = Z_2 - Z_1 \quad (114)$$

$Z_1$  = datum.

$Z_2$  = elevation of the evaporator inlet above datum (taken to be equal to evaporator tube sheet diameter plus 25).

represents the lift head required to move the working fluid to a higher elevation.

The results of piping losses (111), evaporator pressure drop (112), the thermodynamic head (113) and elevation head (114) are equivalent to the ammonia circulation pump system requirements.

$$\Delta P_{\text{pump}} = \Delta P_{\text{PIPE}} + \Delta P_{\text{EVAP}} + \Delta P_{\text{THERMC}} + \Delta P_{\text{ELEVATION}} \quad (115)$$

Using Eq. (93) with ammonia properties, Eq. (115) can be converted to pumping head and finally expressed as pumping power (horsepower).

$$P_{\text{CIRC}} = \frac{\dot{m}_{\text{NH}_3}}{\gamma_p} \left( \frac{gH}{g_c} \right) \quad (116)$$

where  $\dot{m}_{\text{NH}_3}$  = mass flow rate of ammonia determined by Eq. (20) of the previous chapter.

$\gamma_p$  = pump mechanical efficiency (designer input).

Pumping power can then be expressed in terms of megawatts electrical

$$P_{\text{CIRC(MW)}} = \frac{P_{\text{CIRC}}}{\gamma_M} \times \text{CONVERSION FACTOR} \quad (117)$$

where  $\gamma_M$  = pump motor efficiency (designer input).

Because of high pumping head and moderate flow rates, good engineering design would dictate the use of a single suction centrifugal flow type pump.

Using the algorithm developed by Westinghouse Electric Co. [Ref. 15] from data provided by Bingham Pump Division, Portland, Oregon, the cost of the ammonia circulation pump can be expressed as

$$C_{\text{pump}} = \left( \frac{\dot{m}_{\text{NH}_3} v_f}{20100} \right)^{0.64} 1.21 \times 10^5 \quad (118)$$

where  $\dot{m}_{\text{NH}_3}$  = mass flow rate of ammonia (lb<sub>m</sub>/hr)

$v_f$  = specific volume of saturated liquid ammonia  
at state point 1 (ft<sup>3</sup>/lb<sub>m</sub>)

Eq. (118) has been adjusted for current pricing at a 10% annual rate of inflation.

#### 4. Ammonia Re-flux Pump, $P_{\text{RE-FLUX}}$

The function of the re-flux pump is to recycle ammonia droplets which are not evaporated in the heat absorption process. Saturated liquid at approximately the heat exchanger's operating pressure is lifted from the evaporator drain to the ammonia feed inlet, for redistribution as droplets across the evaporator tube bundle. (Drainage mass flow rate is assumed to be equal to 30% of the evaporator inlet feed mass flow rate.)

In order to evaluate these characteristics, the following pump elements will be analyzed:

. Piping losses (friction and minor).

- . Thermodynamic pressure head.
- . Elevation head.

As in the preceding analysis, Reynolds number is used to determine pipe flow characteristics

$$Re_d = \frac{\rho V d_i}{\mu}$$

where  $\rho, \mu$  = saturated liquid properties of ammonia for the average pressure across the evaporator.

$d_i$  = inner pipe diameter (initialized and treated as a design variable by the optimization code).

$V$  = ammonia flow velocity determined from the evaporator drainage mass flow rate assumed equal to 30% of the evaporator inlet feed mass flow rate (assume velocity constant throughout the pipe).

The re-flux pipe friction factor can be determined from Eqs. (85) or (86), and the piping resistance coefficient can be expressed as

$$K = f \frac{L}{d_i} \quad (119)$$

where  $L$  = ammonia re-flux pipe length (designer input).

Once again, considering the resistance coefficient for minor pipe losses assume there are four ninety-degree elbows in the system

$$K = 4 \frac{L}{D} \quad (120)$$

where  $\frac{L}{D}$  = equivalent length in pipe diameters from a standard elbow.

Summing the results of Eqs. (119) and (120), piping losses (friction and minor) can be determined using the Darcy-Weisbach, equation (83)

$$\Delta P_{PIPE} = \rho \left[ f \frac{L}{d_i} + 4 \frac{L}{D} \right] \frac{V^2}{2 g_c} \quad (121)$$

In order to determine the thermodynamic pressure head, the pressure drop across the evaporator for the saturated ammonia liquid must be analyzed. Since the saturated vapor and liquid are in thermodynamic equilibrium, the results of Eq. (112) apply. Therefore

$$\Delta P_{LIQ} = P_3 - P_2$$

Therefore, the thermodynamic pressure head is equal to the pressure drop across the evaporator for the saturated ammonia liquid.

$$\Delta P_{THERMO} = \Delta P_{LIQ} \quad (122)$$

Finally, the elevation head is equal to the elevation of the evaporator feed inlet with respect to datum, the drain outlet.

Therefore,

$$\Delta P_{ELEV} = Z_2 - Z_1 \quad (123)$$

where  $Z_1$  = datum, drain outlet.

$Z_2$  = elevation of the evaporator inlet above datum  
(taken to be equal to the evaporator tube  
sheet diameter plus 10).

The results of piping losses (121), the thermodynamic pressure head (122), and elevation head (123) are equivalent to the ammonia re-flux pump system requirements.

$$\Delta P_{PUMP} = \Delta P_{PIPE} + \Delta P_{ELEVATION} + \Delta P_{THERMO} \quad (124)$$

As before, using Eq. (93), Eq. (124) can be converted to a pump head and finally expressed in terms of pumping power (horsepower).

$$P_{RE-FLUX} = \frac{\dot{m}_R}{\eta_p} \left( \frac{gH}{gc} \right) \quad (125)$$

where  $\dot{m}_R$  = drainage mass flow rate.

$\eta_p$  = pump mechanical efficiency (designer input).

Pumping power can be expressed in terms of megawatts electrical

$$P_{RE-FLUX(MW)} = \frac{P_{RE-FLUX}}{\eta_M} \times \text{CONVERSION FACTOR} \quad (126)$$



where  $\eta_M$  = pump motor efficiency (designer input).

Using the same arguments for the selection of a centrifugal pump, the pump cost algorithm developed by Westinghouse can also be applied to the ammonia re-flux pump.

$$C_{\text{pump}} = \left( \frac{\dot{m}_R v_f}{80100} \right)^{0.64} 1.21 \times 10^5 \quad (127)$$

where  $\dot{m}_R$  = mass flow rate of evaporator drainage ammonia (lb<sub>m</sub>/hr)

$v_f$  = specific volume evaluated at the average evaporator pressure (ft<sup>3</sup>/lb<sub>m</sub>)

Eq. (127) has been adjusted for current pricing at a 10% annual rate of inflation.

##### 5. Parasitic Pump Losses

Parasitic pump losses is the summation of electrical auxiliary pumping requirements (hotel and maintenance loads not included) determined by Eqs. (95), (107), and (126).

$$P_{\text{Loss}} = P_{\text{HP}} + P_{\text{CP}} + P_{\text{CIRC}} + P_{\text{RE-FLUX}} \quad (128)$$

## V. TURBINE AND ELECTRICAL POWER

### A. INTRODUCTION

The turbine generator is one of the critical elements of the OTEC power system. Its energy conversion efficiency and efficiency of design have a major effect on the overall system performance. To illustrate this point, Ref. 16 reported that a three-point change in turbine efficiency from 85 to 88% results in a 3.6% increase in gross power, and a 5% increase in net power developed.

This chapter will describe the analysis to evaluate the expansion turbine thermodynamic properties and generator output. The use of these properties will determine the internal turbine efficiency and outlet quality subject to design and thermodynamic constraints. The relationship between the condenser operating pressure (design variable) and the turbine outlet quality will be used to initialize the heat rejection characteristics of the condenser.

General literature on turbomachinery designed for OTEC closed cycle systems indicates that a turbine having the following characteristics

- . Double flow, axial inflow,
- . Four stages of expansion,
- . Operating at 1800 RPM,

provides the optimum aerodynamic design [Ref. 16]. However, it is not the intent of this thesis to analyze the geometry

and performance parameters of the turbine. Turbine geometry such as

- . Specific speed and specific diameter,
- . Wheel diameter,
- . Rotational speed,
- . Blade height,
- . Blade stresses,

should be treated as a separate systems problem using optimization to improve state-of-the-art design.

Parasitic losses due to the following generator turbine inefficiencies will be evaluated in this section.

- . Generator mechanical and electrical.
- . Turbine mechanical.

As an overview of the turbine-generator analysis, the following major steps of the algorithm are listed in order of their execution:

- . Gross electrical output with no parasitic losses (129).
- . Enthalpy at state point 5 (130).
- . Turbine outlet quality (131).
- . Entropy at state point from a specified outlet quality (132).
- . Quality and enthalpy at state point 5s (133, 134).
- . Internal (adiabatic) turbine efficiency (135).
- . Turbine cost analysis (137).
- . Generator cost analysis (138).

In the following section, the basic steps summarized above will be described in detail.

## B. ANALYSIS OF THE TURBINE AND ELECTRICAL POWER REQUIREMENTS

### 1. Gross Electrical Output and Inefficiency Losses

If the net electrical output required is indicated by (in terms of megawatts), the gross electrical load at the turbine shaft can be expressed as

$$\dot{E}_g = \frac{\dot{E}}{\eta_{TM} \eta_{GEN}} + P_{LOSS} \quad (129)$$

where  $P_{LOSS}$  = parasitic pump losses determined by Eq. (128).

$\eta_{TM}$  = turbine mechanical efficiency (designer input).

$\eta_{GEN}$  = generator mechanical and electrical efficiency (designer input).

The loss of electrical output due to generator-turbine inefficiencies is equal to

$$\dot{E}_{LOSS} = \dot{E} \left( \frac{1}{\eta_{TM} \eta_{GEN}} \right)$$

### 2. Turbine Efficiency

The power developed across the turbine is

$$\dot{E}_g = \dot{m} (h_5 - h_4)$$

where  $\dot{m}$  = mass flow rate of ammonia given by Eq. (48).

$h_4$  = enthalpy at state point 4, Eq. (42).

From this, the enthalpy at state point 5 can be calculated.

If we initialize the operating pressure of the condenser in terms of  $P_5$ , the following relations may be expressed

$$h_{5g} = h_g \Big|_{p_5} \quad h_{5f} = h_f \Big|_{p_5} \quad (130)$$

Therefore, it follows that the turbine outlet quality,  $x_5$ , can be determined from

$$h_5 = h_{5f} + x_5 (h_{5g} - h_{5f}) \quad (131)$$

Having established the moisture separator outlet pressure and temperature, Eqs. (40) and (41), the entropy at state point 4 can be determined for a known separator outlet quality (designer input) using the following relations

$$S_{4f} = S_f \Big|_{T_4} \quad S_{4g} = S_g \Big|_{T_4}$$

$$S_4 = S_{4f} + x_4 (S_{4g} - S_{4f}) \quad (132)$$

For isentropic turbine work,

$$S_4 = S_{5s} \quad (133)$$

the quality at state point 5s may be determined using the following relations

$$S_{5g} = S_g \Big|_{T_5} \quad S_{5f} = S_f \Big|_{T_5}$$

$$S_{5s} = S_{5f} + x_{5s} (S_{5g} - S_{5f}) \quad (134)$$

Having determined the quality at state point 5s, the enthalpy can now be determined.

$$h_{5s} = h_{5f} + X_{5s}(h_{5g} - h_{5f}) \quad (135)$$

Using the results of Eqs. (41), (130), and (132), the internal turbine efficiency (adiabatic) can be determined, expressed by

$$\eta_T = \frac{h_4 - h_5}{h_4 - h_{5s}} \quad (136)$$

To ensure a realistic selection of internal efficiency, the following constraints are attached to the optimization code

- $h_5 < h_{5g}$
- $X_{5s} < X_5$
- $\eta_T \leq 90\%$

### 3. Turbine Cost Analysis

The ammonia turbine cost is based on an algorithm developed by Westinghouse to estimate manufacturing costs [Ref. 15].

$$C_{TURB} = 2.42 \times 10^6 \left( 0.375 + \dot{E}_G / 136000 N_f \right) F_f \quad (137)$$

where  $\dot{E}_G$  = gross electrical output in KW.

$N_f$  = 2 (for a double flow turbine).

$F_f$  = flow price factor (1.0 for single-flow, 1.447 for double-flow).

The above algorithm is valid for the following conditions:

- . Double flow, axial inflow.
- . Multi-stage.
- . Operating at 1800 RPM.

The generator cost will be based on an algorithm developed by TRW from data provided by selected manufacturers,

$$C_{GEN} = (0.023 \dot{E}_G + 0.3) 1.21 \times 10^6 \quad (138)$$

and is valid for the following conditions

- . 1800 RPM rotor speed.
- . power factor 0.8.

Eqs. (137) and (138) have been adjusted for current pricing at a 10% annual rate of inflation.

## VI. CONDENSER

### A. INTRODUCTION

As indicated in the introduction to Chapter III, several heat exchanger concepts have been proposed for the closed-cycle OTEC system, with variations in their design.

The analysis to be presented for the condensing heat exchanger will be based upon the following design characteristics:

- . Single-pass shell and tube heat exchanger.
- . Horizontal/vertical orientation of tubes with an equilateral triangle or square tube profile.
- . Smooth plain-tube configuration (no enhancements).
- . Tube material (titanium or aluminum based on a 30-year life-cycle criterion).
- . Biofouling control based upon an achievable fouling factor.
- . Heat exchanger centerline located on sea surface.

As an overview of the condenser analysis, the following major steps in the algorithm are listed in order of their execution:

- . Initialization of design variables (DV).
  - .. Tube length.
  - .. SW velocity through condenser tubes.
  - .. Outer tube diameter.
  - .. Tube profile pitch ratio.
- . Amount of heat rejection (139).
- . Tubeside bulk temperature (142).



- . Total number of tubes (143).
- . Log mean temperature difference (144).
- . Conductance (146).
- . Number of transfer units (145).
- . Heat exchanger effectiveness (147).
- . Initially assume a value for ammonia heat transfer coefficient (151).
- . Single tube conductance (148).
- . Average heat rejection per tube (152).
- . Film temperature (153).
- . Revised ammonia heat transfer coefficient (154, etc.); iterate with (151).
- . Tube profile, flow parameters across the tube bank (158, etc.).
- . Tube sheet diameter (163).
- . Condenser shellside pressure drop for two-phase flow (166).
- . Revised properties at state point 1 (171, 172); iterate with (21).
- . Overall heat transfer coefficient (173).
- . Total heat transfer surface area (174).
- . Revised condenser tube length (175).
- . Heat exchanger cost analysis.

In the following section, the basic steps summarized above will be described in detail.

## B. ANALYSIS OF THE CONDENSER

### 1. Amount of Heat Rejection, $\dot{Q}$

Using the calculated value for enthalpy at state point 5, equation (131) from the previous chapter, the ideal

values at state point 1, Eq. (21), and the steady-state mass flow rate of ammonia, Eq. (48), the amount of heat rejected by the condenser can be expressed as

$$\dot{Q} = \dot{m}_{NH_3} (h_5 - h_1) \quad (139)$$

## 2. Tubeside Bulk Temperature

As in condenser tubeside Reynolds number, salt water properties will be evaluated at bulk temperature, initially assumed equal to the cold pipe inlet temperature.

Using this premise, the condenser salt water capacity rate can be evaluated

$$\dot{C}_{min} = \dot{m}_{cp} c_{p,sw} \quad (140)$$

where  $c_{p,sw}$  = specific heat of salt water initially evaluated at the cold pipe inlet temperature.

$\dot{m}_{cp}$  = mass flow rate of salt water through the cold pipe previously evaluated by Eq. (107).

Using the results of Eqs. (139) and (140), and the known cold pipe inlet temperature, the condenser salt water outlet temperature may be evaluated from the basic expression

$$\dot{Q} = \dot{C}_{min} (T_{co} - T_{ci}) \quad (141)$$

where  $T_{co}, T_{ci}$  = condenser salt water outlet and inlet temperatures, respectively.

Having determined the condenser salt water outlet temperature, the revised bulk temperature can be expressed as

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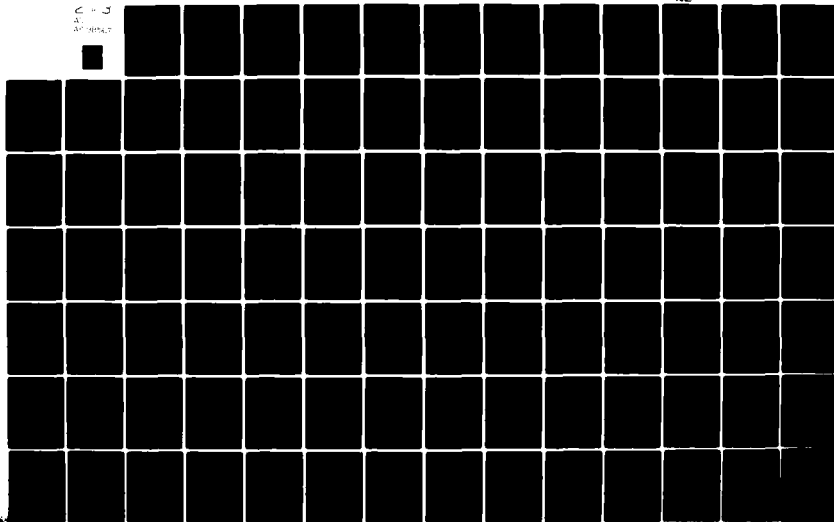
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OPTIMIZATION OF A LOW DELTA T RANKINE POWER SYSTEM. (U)  
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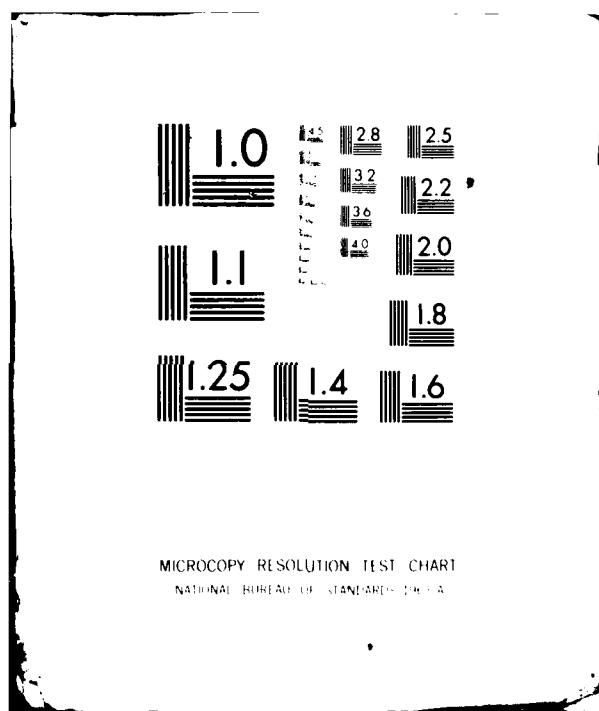
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$$T_B = \frac{T_{c_o} + T_{c_i}}{2} \quad (142)$$

Using the revised condenser bulk temperature and iterating with Eq. (102) corrects the operating temperature for salt water properties which are essential to the analysis.

### 3. Total Number of Condenser Tubes, $N_t$

Since the mass flow rate of salt water through the cold pipe is equivalent to the mass flow rate through the condenser, according to the law of continuity,

$$\dot{m}_{cp} = \dot{m}_{COND}$$

it follows that the number of condenser tubes for a specified tube diameter, can be evaluated using the following expression:

$$\dot{m} = \rho_{sw} \frac{\pi d_i^2}{4} V_t N_t \quad (143)$$

where  $\rho_{sw}$  = average salt water density evaluated at bulk temperature.

$d_i$  = inner tube diameter (initialized and treated as a design variable by the optimization code).

$V_t$  = average salt water velocity through the condenser (initialized and treated as a design variable by the optimization code).

#### 4. Log Mean Temperature Difference, LMTD

Using the result of Eq. (141), the known pipe salt water inlet temperature, and the inlet temperature of ammonia evaluated at state point 5, the LMTD of the condenser may be expressed as

$$LMTD = \frac{T_{c_o} - T_{c_i}}{\ln \left( \frac{T_s - T_{c_i}}{T_s - T_{c_o}} \right)} \quad (144)$$

#### 5. NTU-Effectiveness Relations

The number of transfer units which is a measure of the condenser size can be determined from the basic expression

$$NTU = \frac{U_o A_o}{C_{min}} \quad (145)$$

where the conductance ( $U_o A_o$ ) of the heat exchanger is a function of the heat absorbed and the LMTD.

$$\dot{Q} = (U_o A_o) LMTD \quad (146)$$

The condenser effectiveness can then be expressed as

$$\epsilon = 1 - e^{(-NTU)} \quad (147)$$

for a two-phase flow, regardless of the flow geometry.

## 6. Single-Tube Conductance, $U_o A_o$ .

Using the resistance analysis derived in Chapter III, Section 4 for an initialized tube length

$$L = L_i$$

the heat exchanger conductance for a single tube can be expressed as

$$U_o A_o = \frac{1}{\frac{1}{\eta_i h_{sw} A_i} + \frac{1}{A_i} R_{fsw} + \frac{\ln d_o/d_i}{2\pi K L} + \frac{1}{A_o} R_{fNH_3} + \frac{1}{\eta_o h_{NH_3} A_o}} \quad (148)$$

where  $h_{sw}$  = tubeside heat transfer coefficient.

$R_{fsw}$  = salt water fouling heat transfer resistance.

$K$  = thermal conductivity of the tube material.

$A_o, A_i$  = total outer and inner tube surface areas (including fin and bare tube); tube length is initialized and treated as a design variable by the optimization code).

$R_{fNH_3}$  = ammonia fouling heat transfer resistance

$\eta_o, \eta_i$  = outer and inner total fin efficiency

a. Tubeside Reynolds Number

Since the salt water heat transfer correlation is dependent on tubeside flow, Reynolds number must be evaluated

$$Re_d = \frac{\rho_{sw} V_{sw} d_i}{\mu_{sw}}$$

where  $\rho_{sw}, \mu_{sw}$  = salt water density and viscosity are evaluated for the fluid's bulk temperature.

$d_i, V_{sw}$  = inner diameter and average salt water tube velocity.

Reynolds numbers greater than 2300 will be indicative of turbulent flow [Ref. 3].

b. Salt Water Heat Transfer Coefficient,  $h_{sw}$

Once again the empirical relationship proposed by Sieder and Tate [Ref. 3] will be used for laminar heat transfer in tubes and as defined by

$$Nu_d = 1.86 (Re_d Pr)^{1/3} \left( \frac{d_i}{L} \right)^{1/3} \left( \frac{\mu}{\mu_w} \right)^{0.14}$$

Nusselt and Prandtl numbers are defined as

$$Nu_d = \frac{h_{sw} d_i}{k_{sw}} \quad (149)$$

$$Pr = \frac{c_{p_{sw}} \mu_{sw}}{k_{sw}} \quad (150)$$

where dynamic viscosity, specific heat, and thermal conductivity of salt water are evaluated at the salt water bulk temperature.

The effect of the viscosity ratio in the Sieder-Tate equation is considered negligible, and will hereafter be dropped from the expression. The assumptions and validity condition associated with the Sieder-Tate equation were stated in Chapter III, Section 4, and will not be repeated here.



For fully developed turbulent flow, again the Dittus-Boelter correlation [Ref. 3] was used.

$$Nu_d = 0.023 Re_d^{0.8} Pr^{0.3}$$

Nusselt and Prandtl numbers are previously defined by Eqs. (149) and (150). Assumptions and conditions for validity were stated in Chapter III, Section 4.

c. Salt Water Fouling Heat Transfer Resistance

As indicated previously, it will be assumed that the fouling resistance for tubeside salt water can be maintained at  $.00025 \text{ (hr. ft}^2 \text{ F/BTU)}$ .

d. Ammonia Shellside Heat Transfer Coefficient,  $h_{NH_3}$

Initially,  $h_{NH_3}$  will be assumed

$$h_{NH_3} = 1000 \text{ (BTU/hr. ft}^2 \text{ F)} \quad (151)$$

since its value cannot be directly calculated during this phase of the analysis.

Using the following single-tube thermal resistance

$$R_1 = \frac{1}{\eta_i h_{sw} \pi d_i L}$$

$$R_2 = \frac{1}{\eta_i h_{fsw} \pi d_i L}$$

$$R_3 = \frac{\ln d_o/d_i}{2 \pi K L}$$

$$R_5 = \frac{1}{\eta_o h_{NH_3} \pi d_o L}$$

an initial value for single tube conductance (outer tube surface) may be calculated

$$U_o A_o = \frac{1}{R_1 + R_2 + R_3 + R_5}$$

7. Film Temperature for Property Evaluation,  $T_f$

In order to evaluate the shellside ammonia heat transfer coefficient, working fluid properties must be evaluated at the film temperature.

This can be accomplished by using the results of the single tube conductance, the tube side bulk temperature and the working fluid saturation temperature, expressed in the following equation for single tube heat transfer rate (average).

$$\dot{Q} = U_o A_o (T_5 - T_{BULK}) \quad (152)$$

Again using the resistance analysis as in Chapter III, the shellside wall temperature may be expressed as

$$T_{W2} = T_{BULK} + \dot{Q} (R_1 + R_2 + R_3)$$

Knowing the shellside wall temperature and the free-stream temperature, the film temperature can be derived from their arithmetic mean

$$T_f = \frac{T_{W2} + T_5}{2} \quad (153)$$

For purposes of this calculation, saturated temperature conditions at state point 5 are taken to represent

free-stream conditions, when in fact the two-phase process will experience a pressure drop and a corresponding drop in temperature.

8. Revised Shellside Ammonia Heat Transfer Coefficient,

$h_{NH_3}$

This analysis will include correlations for both horizontal and vertical heat exchangers.

In the horizontal-tubed condenser, Nusselt's correlation was used as a predictor [Refs. 7 and 17],  
for laminar flow

$$\bar{h} = 0.95 \left( \frac{K_f^3 \rho_f^2 g L}{\mu_f w} \right)^{1/3} \quad (154)$$

where  $w$  = estimate of ammonia mass flow rate across each tube.

$K_f, \rho_f, \mu_f$  = properties evaluated at film temperature.

$L$  = tube length (initialized and treated as a design variable by the optimization code).

This correlation is probably conservative, since it does not consider turbulence due to high vapor velocity or splashing of condensate [Ref. 7].

For turbulent flow, Nusselt's correlation is increased by 10% as recommended by Jakob [Ref. 17]

$$\bar{h} = 1.045 \left( \frac{K_f^3 \rho_f g L}{\mu_f w} \right)^{1/3} \quad (155)$$

The laminar-turbulent transition point is defined by a Reynolds number of 2100, where the pseudo-Reynolds number for film-type condensation on horizontal tubes is defined as [Ref. 7]

$$Re = \frac{2\Gamma}{\mu_f}$$

where  $\Gamma$  = mass flow rate of condensate per tube over its length.

In the vertical tubed condenser, both Nusselt's and Kirkbride's correlations were used as predictors [Ref. 7].

For laminar flow, Nusselt's correlation is increased by a factor of 1.28 as recommended by McAdams [Ref. 7]:

$$\bar{h} = 1.28 \left[ 1.47 \left( \frac{\mu_f}{K_f^3 \rho_f^2 g} \right)^{-1/3} \left( \frac{4\Gamma}{\mu_f} \right)^{-1/3} \right] \quad (156)$$

where  $\Gamma$  = mass flow rate of condensate per tube over its diameter.

For turbulent flow, Kirkbride's correlation is applied

$$\bar{h} = 0.0077 \left( \frac{\mu_f^2}{K_f^3 \rho_f^2 g} \right)^{-1/3} \left( \frac{4\Gamma}{\mu_f} \right)^{0.4} \quad (157)$$

The laminar-turbulent transition point is defined by a Reynolds number of 1800, where the pseudo-Reynolds number for film-type condensation on vertical tubes is defined as [Ref. 7]

$$Re = \frac{4\Gamma}{\mu_f}$$

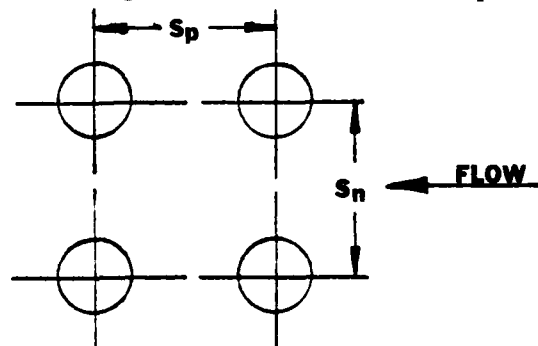
After using the pseudo-Reynolds number to establish the flow in which regime the system is operating, the revised heat transfer coefficient for film-type condensation may be calculated and then iterated with the initial assumption for the shellside heat transfer coefficient, Eq. (151). Once again this will have a convergence effect on variables in which the shellside heat transfer coefficient is a function, moving closer to actual OTEC system operating point characteristics.

9. Tube Profile, Flow across Tube Bank, and Tube Sheet Diameter

Since the condenser tube bundle involves multiple rows of tubes, the geometry of the tube profile arrangement is important to determine the shellside heat transfer coefficient, the tube sheet diameter and the shellside pressure drop associated with the "homogenous" two-phase flow model [Ref. 4].

Using the same arrangements shown in Chapter III, Section 2,

IN-LINE



$$S_n = P_R d_o \quad (158)$$

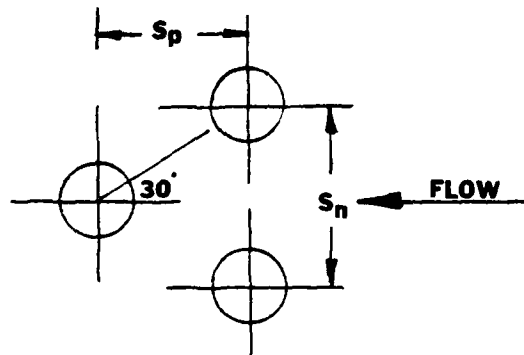
$$A_p = S_n^2 \quad (159)$$

where  $S_n$  = pitch ratio x outer tube diameter.

$P_R$  = pitch ratio (initialized and treated as a design variable for the optimization code).

$A_p$  = tube profile area per tube

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where

$$S_n = 2 P_R d_o \sin 30^\circ \quad (160)$$

$$S_p = P_R d_o \cos 30^\circ \quad (161)$$

$$A_p = S_n S_p \quad (162)$$

the ratio of minimum flow area to the frontal area can be expressed as

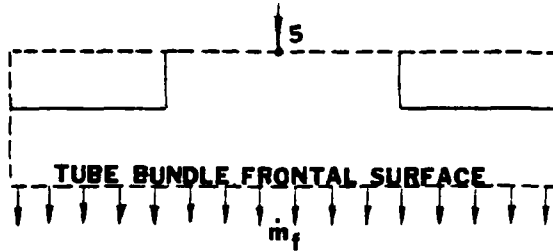
$$\frac{A_{ff}}{A_f} = \frac{S_n - d_o}{S_n} \quad (163)$$

Using the selected tube profile geometry and knowing the number of condenser tubes by Eq. (143), the tube sheet diameter for the condenser design can be evaluated from the following expression

$$N_t A_p = \frac{\pi T_{sd}^2}{4} \quad (164)$$

where  $T_{sd}$  = Tube sheet diameter.

To analyze the shellside ammonia flow velocity, the following control volume is introduced (turbine generator discharge and top portion of the condenser).



Since the mass flow rate remains unchanged across any boundary

$$\dot{m}_5 = \dot{m}_f$$

Furthermore, if we assume the condenser has the capability to evenly distribute vapor across the tube bundle (distribution baffles), the following development applies to the vapor coverage:

Let  $(A_f)_{vap} = A_f \eta$

where  $\eta$  = percent of tube frontal area which is covered by vapor.

$$\dot{m}_5 = \rho_5 A_5 V_5$$

$$\dot{m}_f = \rho_f A_f V_f \eta$$

where  $A_5$  = condenser inlet cross-sectional area.

$V_5$  = turbine discharge ammonia velocity.

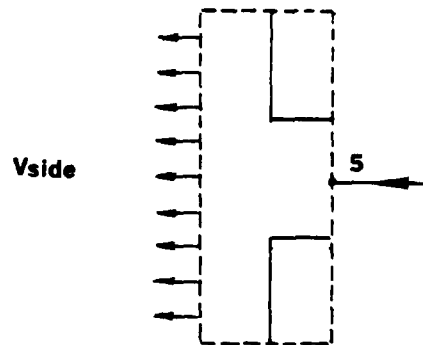
Therefore

$$V_f = \frac{\rho_5 A_5}{\rho_f A_f \eta} V_5$$

If  $\eta = \rho_5 A_5 / A_f \rho_f$ , it follows that the turbine discharge velocity is equal to the average velocity of ammonia at the tube frontal area boundary. A determination of the distribution fraction  $\eta$  requires a detailed knowledge of the design of the turbine/condenser interface. In the absence of this information it is assumed that

$$V_f = V_5$$

A similar argument could be presented for a vertical tubed condenser where turbine discharge is admitted to a distribution ring that bands the condenser tube bank. Exhaust vapor would travel radially through the tube bundle and then collect at the bottom after vertical film-condensation.



Again, in the absence of a detailed design, it is assumed that

$$V_{SIDE} = V_5$$



Considering the minimum free-flow area for a horizontal tubed condenser,  $A_{ff}$  can be derived using Eq. (163) and the projected frontal area.

$$A_f = T_{SD} L_t$$

$$A_{ff} = A_f \left( \frac{S_n - d_o}{S_n} \right) \quad (165)$$

where  $A_f$  = the flow frontal area.

$L_t$  = tube length.

For vertical condensers

$$A_f = \pi T_{SD} \times \text{FRONTAL LENGTH OF VAPOR INLET FLOW}$$

Using the previously calculated value of the ammonia flow rate and Eq. (165), mass velocity for the minimum free flow area can be expressed as

$$G = \frac{\dot{m}_4}{A_{ff}} \quad (166)$$

#### 10. Pressure Drop of Two-Phase Flow across a Bank of Tubes, $\Delta P$

The pressure drop in the two-phase flow condensing heat exchanger will be determined using the homogeneous model introduced in Chapter III. The model will consist of three components -- friction loss, momentum change, and elevation pressure drop arising from the effects of gravity.

The local pressure drop for a two-phase flow may be expressed as

$$\Delta P_{COND} = \Delta P_{FRICTION} + \Delta P_{MOMENTUM} + \Delta P_{ELEVATION} \quad (167)$$

For a given channel length,  $L_c$ , the pressure drop components can be expressed by

$$\Delta P_{FRICTION} = \frac{f G^2 \bar{v}}{D_e 2 g_c} L_c \quad (168)$$

$$\Delta P_{MOMENTUM} = \frac{G^2 \bar{v}}{g_c} \quad (169)$$

$$\Delta P_{ELEVATION} = \frac{g}{\bar{v} g_c} L_c \quad (170)$$

where  $f$  = single phase friction factor by Jakob expressed in Eq. (35) or (36).

$G$  = mass flow velocity determined from Eq. (166).

$L_c$  = channel flow length, defined for horizontal tubed condensers as  $L_c = T_{SD}$  (tube sheet diameter) and for vertical tubed condensers as  $L_c = L_t$  (tube length).

$D_e$  = equivalent diameter of flow channel, defined by

$$D_e = P_R d_o - d_o$$

$\bar{v}$  = mean specific volume defined by

$$\bar{v} = v_f \left[ 1 + \frac{x}{v_f} (v_g - v_f) \right]$$

where  $X$  = quality of mixture (state point 5).

$v_f$  = specific volume of liquid (state point 1)

$v_g$  = specific volume of vapor (state point 5).

All components of the pressure drop model Eqs. (168, 169, and 170) can be determined using the preceding information.

#### 11. Revised Properties at State Point 1

Since Eq. (167) represents the pressure drop across the condenser shellside, the actual pressure at state point 1 or condenser outlet may be determined from

$$P_1(\text{NEW}) = P_1 - \Delta P_{\text{COND}} \quad (171)$$

where  $P_1$  is previously described as the condenser operating pressure for the ideal cycle.

Operating on the saturated liquid line on the Temperature-Entropy diagram, the following properties are defined:

$$h_1(\text{NEW}) = h_f \big|_{P_1(\text{NEW})} \quad T_1(\text{NEW}) = T_{\text{SAT}} \big|_{P_1(\text{NEW})} \quad (172)$$

The subscript (NEW) representing a revised property will hereafter be dropped from the expression in Eq. (172).

Until now, we assumed the condenser outlet temperature and pressure were designed to operate as an ideal system, without a pressure drop. Therefore, using the revised temperature at state point 1 and iterating over the range from Eq. (21) until an acceptable convergence criterion is achieved, all the preceding variables as function of  $T_1$

will be reevaluated to complete the closed-loop cycle of the simulated OTEC power system.

## 12. Overall Heat Transfer Coefficient, $U_o$

The quantity "U" represents a measure of the total thermal resistances in the flow path. Therefore, using the tube conductance expressed in Eq. (148) which is divided by the outer heat transfer surface area of a single tube, the overall heat transfer coefficient for the condenser can be determined.

The thermal resistances are now expressed as

$$R_1 = \frac{d_o}{\eta_i h_{sw} d_i}$$

$$R_2 = \frac{d_o}{\eta_i h_{fsw} d_i}$$

$$R_3 = \frac{d_o \ln d_o / d_i}{2K}$$

$$R_5 = \frac{1}{\eta_o h_{NH_3}}$$

and the overall heat transfer coefficient for the condenser may be calculated using

$$U_o = \frac{1}{R_1 + R_2 + R_3 + R_5} \quad (173)$$

### 13. Total Condenser Heat Transfer Surface Area, $A_t$

Having determined the corrected number of condenser transfer units (145), salt water capacity rate (140) and overall heat transfer rate (173), the total condenser heat transfer area can be calculated from the NTU expression

$$NTU = \frac{U_o A_t}{C_{min}} \quad (174)$$

### 14. Revised Condenser Tube Length

Using the total heat transfer surface area calculated from Eq. (174) and the total number of condenser tubes (143), the revised condenser tube length can be determined from the basic expression

$$A_t = N_t \pi d_o L_t (REVISED) \quad (175)$$

At this time, it is necessary to iterate the condenser design until the two values (initial and revised) of the tube length converge. This iteration may be accomplished by the COPES routine if the following constraint is defined

$$L_{DIFF} = L_t (REVISED) - L_t (INITIAL)$$

Minimization of this difference will cause continual adjustment of the required tube length, already treated as a design variable by the optimization code.

## 15. Condenser Heat Exchanger Cost Analysis

As indicated in Chapter III, TRW developed sets of equations to represent the costs of various heat exchanger component parts for shell diameters ranging from 10-35 feet and 35-50 feet [Ref. 9].

The following are the TRW component cost equations for the condensing heat exchanger. Prior equation reference numbers will be substituted where equalities exist with the evaporative heat exchanger component cost expressions.

for tube sheet diameter 10-35 feet

. Drilling time/tube sheet thickness. (58)

. Thickness of the tube sheet. (59)

. Tube sheet labor cost. (60)

. Tube sheet material cost. (61)

. Tube installation cost. (62)

. Heat exchanger drill cost. (63)

. Ammonia distribution plate and baffles cost.

$$C_{DPB} = 1.539 \times 10^{-2} t_d N_t T_{SD}^{2.0} \quad (176)$$

. Bustle, flanges, channels and flow plate cost.

$$C_{BFCF} = 1185.286 T_{SD}^{2.0} \quad (177)$$

. Tube material cost.

$$C_{TM} = (C_1 L_t + C_2) N_t d_o / 1.5 \quad (178)$$

where  $C_1$  = curve fit of tube material cost  
per foot.

$C_2$  = tube machining cost if required.

. Heat exchanger header cost. (67)

. Water inlet, nozzles and support cost.

$$C_{WINS} = 10106.475 T_{SD} \quad (179)$$

. Tube welding costs (Titanium tubes). (69)

The sum of the preceding costs would equal the cost to fabricate one OTEC condenser with a tube sheet diameter of 10-35 feet (all the preceding component costs have been adjusted for current pricing at a 10% annual rate of inflation).

If our analysis is based on a 30-year life-cycle criterion, no adjustments are necessary to any component cost equation if titanium tubing is selected. However, using Al 5052-0, the expense of retubing must be considered to meet the 30-year life-cycle criterion, as in the case of the evaporation. For convenience, and possible subsequent modification, these considerations are repeated here.

Based upon the utility of Al 5052-0, two complete condenser retubings will be required to meet the basic 30-year criterion. This implies Eqs. (62) and (178) must be modified to reflect the costs of retubing at the 10 and 20 year point in the cycle.

. Aluminum tube installation cost.

$$C_{ATI} = C_{TI} [1 + (1+i)^{10} + (1+i)^{20}] \quad (180)$$

where  $i$  = projected annual inflationary  
rate (input by customer).

. Aluminum tube material cost.

$$C_{ATM} = C_{TM} [1 + (1+i)^{10} + (1+i)^{20}] \quad (181)$$

for tube sheet diameter 35-50 feet.

. Drilling time/tube sheet thickness (58)

. Thickness of the tube sheet. (59)

. Tube sheet labor and material costs  
(titanium). (72, 73)

. Tube sheet labor and material costs  
(aluminum). (74, 75)

. Tube installation cost. (76)

. Tube material cost. (178)

. Heat exchanger shell cost. (77)

. Ammonia distribution plate and  
baffles cost.

$$C_{DPB} = 9.825 N_t^{0.479} t_d \quad (182)$$

. Bustle, flanges, channels and  
flow plate.

$$C_{BFCF} = 382.824 T_{SD}^{2.184} \quad (183)$$

. Heat exchange head cost.

$$C_{HxH} = 939.62 T_{SD}^{1.43} \quad (184)$$



. Water inlet, nozzles, and supporters  
cost.

$$C_{WINS} = 7453.6 T_{SD}^{1.056} \quad (185)$$

. Tube welding cost (titanium tubes). (82)

As indicated previously, the cost to fabricate one OTEC condenser with a tube sheet diameter of 35 to 50 feet is equal to the sum of component costs (note, all the preceding component costs have been adjusted for current pricing at a 10% annual inflation rate).

For an analysis based on a 30-year life-cycle criterion, the additional costs for replacing aluminum tubing must be considered and Eqs. (180) and (181) apply.

## VII. NUMERICAL OPTIMIZATION

### A. INTRODUCTION

Nearly all design processes attempt the minimization or maximization of some parameter or design objective. For the design to be acceptable, it must satisfy a set of constraints which impose limits or bounds on design parameters.

For the stated problem a computer program can be written to perform the basic analysis of the proposed design. If any parameters fall outside the prescribed bounds, the design engineer changes the parameters and re-runs the program. In effect, the computer code provides the analysis with the engineering making the actual design decisions.

A logical extension to the computer-aided approach is a fully automated design, where the computer also makes the actual design decisions and performs trade-off studies. The COPES program provides this automated design and trade-off capability by the use of the optimization program COPES/CONMIN [Ref. 18]. COPES is an acronym for Control Program for Engineering Synthesis, and CONMIN is an acronym for CONstrained function MINimization. Subsequently, a FORTRAN analysis program simulating a closed-cycle OTEC power system can be coupled to the COPES program for automated design, using some basic programming guidelines [Ref. 18].

## B. COPES/CONMIN

There are many numerical optimization schemes available to the engineer. Methods employed by these schemes fall into four basic categories: random search, sequential unconstrained minimization, optimality criteria, and direct constrained optimization. The optimization program, selected for automated design analysis of the simulated OTEC power system, is based upon direct constrained optimization.

Before any discussion of the optimization technique, basic definitions are summarized for convenient reference [Ref. 19]:

- . Design variables - those parameters which the optimization program is permitted to change in order to improve the program.
- . Objective function - the parameter which is to be minimized or maximized during the optimization process.
- . Inequality constraint - one-sided conditions which must be satisfied for an acceptable design.
- . Equality constraint - condition which must be equaled for the design to be acceptable.

. Side constraints - upper and lower bounds in a design variable.

Assuming that the FORTRAN analysis program has been developed and a particular objective function has been selected, the general optimization problem can be stated as [Ref. 20]:

Find the vector of design variables,  $\bar{X}$ , to

$$\text{Minimize } F(\bar{X}) \quad (186)$$

Subject to the constraints:

$$G_j(\bar{X}) \leq 0 \quad j=1, NCON \quad (187)$$

$$H_j(\bar{X}) = 0 \quad j=1, NEQ \quad (188)$$

$$VLB_i \leq \bar{X}_i \leq VUB_i \quad i=1, NDV \quad (189)$$

where  $\bar{X}$  = the vector containing the set of independent design variables.

$F(\bar{X})$  = the objective function to be minimized.

$G_j(\bar{X})$  = inequality constraint (NCON is the number of such constraints).

$H_j(\bar{X})$  = equality constraint (NEQ is the number of such constraints).

$VLB_i/VUB_i$  = lower and upper bounds, respectively, on the design variables.

If all inequalities of Eqs. (187) and (189) are satisfied, the design is said to be feasible if any constraint is not satisfied, the design is infeasible. If the objective function

is a minimum and the design is feasible, it is said to be the optimal design.

In order to start the optimization algorithm, the initial set of design variables,  $\bar{X}$ , must be specified. It is desirable, but not essential, that the initial design variables provide a feasible solution. The optimization algorithm will then proceed in an iterative fashion using the following relationship

$$\bar{X}^{q+1} = \bar{X}^q + \alpha * \bar{S}^q$$

where  $q$  = the iteration number.

$\alpha$  = scalar quantity which defines the move in the search direction.

$\bar{S}$  = vector search direction which will reduce the objective function (useable direction) without violating constraints (feasible direction).

To solve this problem, the optimization program COPES/CONMIN is used [Ref. 18]. CONMIN uses the Fletcher-Reeves algorithm for locally unconstrained problems [Ref. 20] and Zoutendijk's method of feasible directions (modified to improve efficiency and reliability and to deal with designs which do not initially satisfy all the constraints) for locally constrained problems [Ref. 21].

However, CONMIN does not handle equality constraints directly, but rather by means of penalty parameters. To achieve this, the objective function is augmented as follows

[Ref. 19]:

$$F'(\bar{x}) = F(\bar{x}) - K \sum_{j=1}^{NEQ} H_j \quad (190)$$

and the equality condition of Eq. (188) is treated as an inequality constraint

$$H_j(\bar{x}) \leq 0 \quad j=1, NEQ$$

The penalty function approach effectively satisfies the equality constraint while maintaining the rapid convergence characteristics of the CONMIN program.

The numerical optimization problems of equations (186) through (190) are very general, allowing for any number of design variables and constraints. In assessing the value of optimization, the automated design provides a very attractive approach to numerical optimization; however, there are both advantages and limitations to these techniques [Ref. 20].

Advantages:

- . Reduction in design time.
- . Systematic design procedure.
- . Applicable to a wide variety of design variables and constraints.
- . Virtually always yields some design improvement.
- . Not biased by engineering experience.
- . Requires a minimal amount of man-machine interface.

Limitations:

- . Computer times may increase dramatically as the number of design variables increases. A practical limit imposed by the current state of the art for most problems is 30 design variables.

- . Optimization techniques have no stored experience to draw upon; the validity of the result is limited to the validity of the analysis program.
- . The results of the optimization are as correct as the analysis program is theoretically precise.
- . Optimization algorithms used here cannot deal with discontinuous functions.
- . The optimization program will not always obtain a global design optimum and may require restarting from several different points to acquire reasonable assurance of obtaining the global optimum.
- . The analysis program must be properly structured to couple with the COPES/CONMIN optimization code.

#### C. DESIGNATED DESIGN VARIABLES, CONSTRAINTS AND OBJECTIVE FUNCTION

To assist in the interpretation of the enclosed OTEC power system FORTRAN analysis, the following summary identifies the design variables, constraint functions and objective function used in the analysis and subsequently operated upon by the COPES/CONMIN optimization code. These parameters are all contained in a labeled COMMON block in the computer code, referred to here as "GLOBAL COMMON." Specific GLOBAL COMMON location numbers and upper/lower bounds for operating parameters summarized below can be located in Appendix C.

##### Design Variables

- . Inner cold pipe diameter
- . Inner hot pipe diameter
- . Inner ammonia circ pipe diameter
- . Inner ammonia re-flux pipe diameter

- . Evaporator operating pressure
- . Condenser operating pressure
- . Outer condenser tube diameter
- . Outer evaporator tube diameter
- . Evaporator tube length
- . Condenser tube length
- . Condenser tube salt water velocity
- . Cold pipe salt water velocity
- . Evaporator tube salt water velocity
- . Hot pipe salt water velocity
- . Evaporator tube profile pitch ratio
- . Condenser tube profile pitch ratio

#### Constraint Functions

- . Operating system pressure ratio
- . Upper temperature bound of ammonia
- . Lower temperature bound of ammonia
- . Satisfactory enthalpy at state point 5
- . Satisfactory quality at state point 5
- . Satisfactory condenser tube length
- . Internal turbine efficiency
- . Evaporator tube sheet diameter
- . Condenser tube sheet diameter

#### Objective Function

- . Cost of major power system components



## VIII. CONCLUSIONS AND RECOMMENDATIONS

### A. CONCLUSIONS

1. The use of an analysis code for OTEC power systems coupled to COPES/CONMIN optimization code provides a powerful tool to design an optimum power system for the desired net electrical output, measured against the objective function. Such a design could permit construction of higher capacity systems using the optimized modules as substations of the total power plant.

2. The analysis code coupled to COPES/CONMIN provides an excellent vehicle to evaluate proposed designs relative to a true optimum. Tables 1 through 4 illustrate the result of preliminary calculations using the analysis code with an objective function to minimize system cost. From these, the following conclusions can be drawn concerning horizontally oriented aluminum (Al-5052) and titanium-tubed heat exchanger power systems:

a. The cost/KW output is nearly constant over the range of optimum designs for both titanium and aluminum tube heat exchangers.

b. During testing for feasible plant designs in increments of 5 MW (net) electrical output, it was observed that a higher megawatt output plant could be achieved with titanium-tubed heat exchangers than for aluminum (Al-5052). For titanium-tubed heat exchangers, a 25 MW (net) power

system is a feasible design; however, aluminum-tubed systems could not provide a feasible design for the same output. Titanium tubed plants failed to produce a feasible design for a 30 MW (net) output power system. In both cases of infeasible design, the constraint which was consistently violated was turbine internal efficiency, set at 90% for current state-of-the-art design.

c. The energy conversion and efficiency of design of a turbine-generator has a major effect on the overall system performance as indicated in paragraph b above.

d. The cost/KW output for titanium-tubed heat exchangers is one third the cost/KW output for aluminum-tubed heat exchangers using a 30-year life-cycle criterion, with a 10% annual inflation rate and retubing at 10 and 20 year marks with AL-5052 tubing.

e. Aluminum-tubed heat exchangers have larger tube bundle volumes, with volumetric differences between aluminum and titanium varying from 26.1 to 11.8% for evaporators and 23.2 to 7.4% for condensers over the range of net power levels considered. In both cases volumetric differences diminish as the system's net electrical output increases to 20 megawatts.

f. COPES/CONMIN has provided optimum designs for each incremental output power level. By manipulating the specified design variables, subject to imposed constraints, COPES/CONMIN has created designs whose geometry and operating

parameters cannot be scaled on the basis of net power output (10 MW). Therefore, designs for component geometry at increasing power levels based upon such simplistic scaling criteria will not achieve an optimum design with respect to the cost objective function.

#### B. RECOMMENDATIONS

1. Evaluate additional objective functions including:

- a. Minimize heat exchanger volumes.
- b. Minimize parasitic power losses.
- c. Maximize thermodynamic efficiency.
- d. Maximize net electrical output.

2. Perform a sensitivity analysis on power system design variables to evaluate their influence on component and system performance. This allows the designer to prioritize system components which can provide improvement in the objective function for a corresponding improvement in component design.

3. Considerable uncertainties are associated with the expressions used to estimate component performances (two-phase pressure drops, film coefficients, etc.). The code should be tested to determine the sensitivity of system design to these uncertainties.

4. Expand the code to include the use of enhanced heat transfer techniques and evaluate the influence of increased piping friction factors on pumping power requirements.

5. Evaluate proposed OTEC designs using proposed system parameter inputs, comparing both the basic analysis and the optimization output.

6. Select other analytical expressions for heat transfer coefficients to validate the performance and output of the existing code.

7. Evaluate the effect of a smaller thermal difference seen by the power system and its influence on a feasible design for a specific net electrical output.

8. Evaluate the cost aspects of using variable-pitch pumps versus fixed-blade for a variable thermal gradient environment.

9. Evaluate and verify the influence of incremental improvements (percent) in turbine internal/adiabatic efficiency with respect to gross and net electrical outputs and compare with the results reported in Ref. 16.

TABLE 1: OTEC Power System Comparisons (Titanium Tubed Heat Exchangers)

EVAPORATOR	10 MW	15 MW	20 MW	25 MW
HT ABSORB (BTU/HR)	1.44 E09	2.01 E09	2.80 E09	3.46 E09
SW FLOW (LB <sub>m</sub> /HR)	2.27 E08	3.34 E08	4.14 E08	4.86 E08
NH <sub>3</sub> FLOW (LB <sub>m</sub> /HR)	2.73 E06	3.78 E06	5.29 E06	6.53 E06
OPER PRESS (LB <sub>f</sub> /IN <sup>2</sup> )	129.	130.0	127.7	127.1
OVL HT COEF (BTU/ HR·FT <sup>2</sup> ·F)	623.19	612.97	601.47	595.73
HT SURFACE (FT <sup>2</sup> )	387,598	572,405	743,093	922,075
TUBE OUTER DIA (IN)	0.947	0.952	0.945	0.929
TUBE WALL THICK (IN)	0.025	0.025	0.025	0.025
TUBE PROFILE - STAGGERED EQUILATERAL TRIANGLE				
PITCH RATIO	1.4	1.4	1.47	1.52
TUBE LENGTH (FT)	43.66	42.18	42.27	42.56
TUBE SHEET DIA (FT)	21.96	27.21	32.45	36.76
TOT NR OF TUBES	35,806	54,449	71,034	89,105

TABLE 1. OTEC Power System Comparisons (Continued)

CONDENSER	10 MW	15 MW	20 MW	25 MW
HT REJECT (BTU/HR)	1.39 E09	1.94 E09	2.71 E09	3.34 E09
SW FLOW (LB <sub>m</sub> /HR)	2.23 E08	3.35 E08	4.72 E08	5.69 E08
NH <sub>3</sub> FLOW (LB <sub>m</sub> /HR)	2.73 E06	3.79 E06	5.29 E06	6.53 E08
OPER PRESS (LB <sub>f</sub> /IN <sup>2</sup> )	88.16	88.15	87.46	87.49
OVL HT COEF (BTU/ HR·FT <sup>2</sup> ·F)	454.4	446.8	438.4	435.7
HT SURFACE (FT <sup>2</sup> )	552,314	762,190	1,168,239	1,483,762
TUBE OUTER DIA (IN)	0.935	0.972	0.957	0.940
TUBE WALL THICK (IN)	0.025	0.025	0.025	0.025
TUBE PROFILE - STAGGERED EQUILATERAL TRIANGLE				
PITCH RATIO	1.4	1.4	1.48	1.51
TUBE LENGTH (FT)	58.57	57.42	58.32	59.09
TUBE SHEET DIA. (FT)	22.48	27.194	35.16	39.734
TOT NR OF TUBES	38,524	52,179	79,956	102,054

TABLE 1: OTEC Power System Comparisons (Continued)

PIPING SYSTEMS	10 MW	15 MW	20 MW	25 MW
SW HOT PIPE (300 FT LENGTH)				
INNER DIA (FT)	17.20	20.08	21.86	23.30
SW VEL (FT/SEC)	4.26	4.6	4.80	4.97
PRESS DROP ( $\text{LB}_f/\text{IN}^2$ )	0.280	0.322	0.348	0.371
SW COLD PIPE (3000 FT LENGTH)				
INNER DIA (FT)	16.1	18.62	21.35	22.97
SW VEL (FT/SEC)	4.94	5.33	5.72	5.95
PRESS DROP ( $\text{LB}_f/\text{IN}^2$ )	0.49	0.508	0.526	0.539
$\text{NH}_3$ CIRC PIPE (150 FT LENGTH)				
INNER DIA (FT)	2.0	2.0	2.0	2.0
PRESS DROP ( $\text{LB}_f/\text{IN}^2$ )	8.46	9.87	11.33	12.53
$\text{NH}_3$ RE-FLUX PIPE (50 FT LENGTH)				
INNER DIA (FT)	2.0	2.0	2.0	2.0
PRESS DROP ( $\text{LB}_f/\text{IN}^2$ )	8.46	9.87	11.33	12.53

TABLE 1: OTEC Power System Comparisons (Continued)

PUMP SYSTEMS	10 MW	15 MW	20 MW	25 MW
EVAP SW PUMP (EFFICIENCY 85 PCT)				
HEAD (FT)	11.0	9.89	9.42	9.26
CAPACITY (GAL/MIN)	444,338	653,260	808,625	930,352
COND SW PUMP (EFFICIENCY 85 PCT)				
HEAD (FT)	22.23	20.75	20.04	20.03
CAPACITY (GAL/MIN)	452,026	652,119	920,116	1,107,382
NH <sub>3</sub> CIRC PUMP (EFFICIENCY 75 PCT)				
HEAD (FT)	200.9	217.33	214.93	220.52
CAPACITY (GAL/MIN)	8709.3	12,101.7	16,880.2	20,586.8
NH <sub>3</sub> RE-FLUX PUMP (EFFICIENCY 75 PCT)				
HEAD (FT)	32.4	38.02	43.53	48.13
CAPACITY (GAL/MIN)	2684.3	3732.4	5201.4	6424.3



TABLE 1: OTEC Power System Comparisons (Continued)

EFFICIENCY OF OPERATION	10 MW	15 MW	20 MW	25 MW
TURBINE-GENERATOR (TURB MECH 99.8 PCT, GEN MECH AND ELECT 96.6 PCT)				
TURB INTERNAL (PCT)	87.16	89.88	88.22	89.80
OUTLET QUALITY (PCT)	96.6	96.77	96.9	96.9
POWER REQUIREMENTS (MEGAWATTS)				
TURB EFFIC LOSSES	0.373	0.559	0.745	0.932
EVAP SW PUMP	1.131	1.496	1.762	2.034
COND SW PUMP	2.334	3.143	4.283	5.151
NH <sub>3</sub> CIRC PUMP	0.281	0.412	0.583	0.738
NH <sub>3</sub> RE-FLUX PUMP	0.014	0.022	0.035	0.048
TURB-GEN GROSS	14.132	20.633	27.409	33.904
PCT PARASITIC POWER	26.6	24.6	24.31	23.51
THERMO CYCLE EFFIC (PCT)	2.45	2.65	2.53	2.56

TABLE 1: OTEC Power System Comparisons (Continued)

COMPONENT COSTS (\$)	10 MW	15 MW	20 MW	25 MW
EVAPORATOR	5,134,322.	8,223,672.	11,501,680.	13,026,794.
CONDENSER	5,952,658.	8,667,154.	13,011,939.	16,558,713.
GEN-TURBINE	1,495,085.	1,578,776.	1,666,011.	1,749,636.
GENERATOR	756,287.	937,205.	1,125,782.	1,306,554.
EVAP SW PUMP	463,736.	653,333.	794,355.	922,944.
COND SW PUMP	470,714.	652,298.	895,505.	1,065,449.
NH <sub>3</sub> CIRC PUMP	110,849.	136,825.	169,303.	193,848.
NH <sub>3</sub> RE-FLUX PUMP	52,190.	64,449.	79,701.	91,233.
OPTIMUM COST (\$)	14,383,650.	20,849,216.	29,164,560.	34,823,904.
COST/KW (NET) OUTPUT (\$/KW)	1438.36	1389.95	1458.23	1392.96

TABLE 2: QTEC Power System Comparisons (Aluminum Tubed Heat Exchanger)

EVAPORATOR	10 MW	15 MW	20 MW	25 MW
HT ABSORB (B U/HR)	1.51 E09	1.97 E09	2.80 E09	
SW FLOW (LB <sub>m</sub> /HR)	2.53 E08	3.42 E08	3.89 E08	
NH <sub>3</sub> FLOW (LB <sub>m</sub> /HR)	2.86 E06	3.72 E06	5.29 E06	
OPER PRESS (LB <sub>f</sub> /IN <sup>2</sup> )	129.57	131.98	128.01	
OVL HT COEF (BTU/ HR·FT <sup>2</sup> ·F)	646.7	643.2	625.5	
HT SURFACE (FT <sup>2</sup> )	391,370	610,261	777,553	
TUBE OUTER DIA (IN)	1.221	1.046	0.982	
TUBE WALL THICK (IN)	0.065	0.065	0.065	
TUBE PROFILE - STAGGERED EQUILATERAL TRIANGLE				
PITCH RATIO	1.4	1.46	1.6	
TUBE LENGTH (FT)	47.68	42.42	41.63	
TUBE SHEET DIA (FT)	23.97	30.73	34.68	
TOT NR OF TUBES	25,669	52,512	72,662	

INFEASIBLE DESIGN

TABLE 2: OTEC Power System Comparisons (Continued)

CONDENSER	10 MW	15 MW	20 MW	25 MW
HT REJECT (BTU/HR)	1.47 E09	1.89 E09	2.71 E09	
SW FLOW (LB <sub>m</sub> /HR)	2.44 E08	3.04 E08	4.38 E08	
NH <sub>3</sub> FLOW (LB <sub>m</sub> /HR)	2.86 E06	3.72 E06	5.30 E06	
OPER PRESS (LB <sub>f</sub> /IN <sup>2</sup> )	88.52	89.18	87.85	
OVL HT COEF (BTU/ HR·FT <sup>2</sup> ·F)	454.57	453.54	446.48	
HT SURFACE (FT <sup>2</sup> )	554,011	690,395	1,173,216	
TUBE OUTER DIA (IN)	1.176	1.092	1.001	
TUBE WALL THICK (IN)	0.065	0.065	0.065	
TUBE PROFILE - STAGGERED EQUILATERAL TRIANGLE				
PITCH RATIO	1.4	1.46	1.5	
TUBE LENGTH (FT)	64.04	57.46	57.24	
TUBE SHEET DIA (FT)	24.15	28.61	36.82	
TOT NR OF TUBES	28,091	42,035	78,198	

INFEASIBLE DESIGN

TABLE 2: OTEC Power System Comparisons (Continued)

PIPING SYSTEMS	10 MW	15 MW	20 MW	25 MW
SW HOT PIPE (300 FT LENGTH)				
INNER DIA (FT)	17.89	20.25	21.33	
SW VEL (FT/SEC)	4.38	4.63	4.74	
PRESS DROP ( $\text{LB}_f/\text{IN}^2$ )	0.295	0.326	0.341	
SW COLD PIPE (3000 FT LENGTH)				
INNER DIA (FT)	16.53	17.93	20.7	
SW VEL (FT/SEC)	4.93	5.23	5.64	
PRESS DROP ( $\text{LB}_f/\text{IN}^2$ )	0.478	0.504	0.523	
$\text{NH}_3$ CIRC PIPE (150 FT LENGTH)				
INNER DIA (FT)	1.99	2.0	2.0	
PRESS DROP ( $\text{LB}_f/\text{IN}^2$ )	13.78	15.94	17.89	
$\text{NH}_3$ RE-FLUX PIPE (50 FT LENGTH)				
INNER DIA (FT)	2.0	2.0	2.0	
PRESS DROP ( $\text{LB}_f/\text{IN}^2$ )	8.996	10.788	11.916	

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TABLE 2: OTEC Power System Comparisons (Continued)

PUMP SYSTEMS	10 MW	15 MW	20 MW	25 MW
EVAP SW PUMP (EFFICIENCY 85 PCT)				
HEAD (FT)	10.42	10.27	10.06	
CAPACITY (GAL/MIN)	494,239	669,211	760,748	
COND SW PUMP (EFFICIENCY 85 PCT)				
HEAD (FT)	21.32	20.80	21.05	
CAPACITY (GAL/MIN)	474,918	592,656	851,978	
NH <sub>3</sub> CIRC PUMP (EFFICIENCY 75 PCT)				
HEAD (FT)	203.5	218.32	216.63	
CAPACITY (GAL/MIN)	9144.3	11,895.6	16,911.6	
NH <sub>3</sub> RE-FLUX PUMP (EFFICIENCY 75 PCT)				
HEAD (FT)	34.33	41.27	45.66	
CAPACITY (GAL/MIN)	2818.4	3669.9	5210.5	

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TABLE 2: OTEC Power System Comparisons (Continued)

EFFICIENCY OF OPERATION	10 MW	15 MW	20 MW	25 MW
TURBINE-GENERATOR (TURB MECH 99.8 PCT, GEN MECH AND ELECT 96.6 PCT)				
TURB INTERNAL (PCT)	83.37	89.76	88.05	
OUTLET QUALITY (PCT)	97.06	96.73	96.93	
POWER REQUIREMENTS (MEGAWATTS)				
TURB EFFIC LOSSES	0.373	0.559	0.745	
EVAP SW PUMP	1.191	1.590	1.770	
COND SW PUMP	2.351	2.863	4.166	
NH <sub>3</sub> CIRC PUMP	0.299	0.416	0.588	
NH <sub>3</sub> RE-FLUX PUMP	0.015	0.024	0.037	
TURB-GEN GROSS	14.229	20.452	27.306	
PCT PARASITIC POWER	27.10	23.92	24.03	
THERMO CYCLE EFFIC (PCT)	2.34	2.70	2.53	

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TABLE 2: OTEC POWER SYSTEM COMPARISONS (CONTINUED)

COMPONENT COSTS (\$)	10 MW	15 MW	20 MW	25 MW
EVAPORATOR	20,848,960.	34,297,296.	44,203,408.	
CONDENSER	24,633,360.	31,812,640.	55,404,448.	
GEN-TURBINE	1,496,339.	1,576,449.	1,664,688.	
GENERATOR	758,998.	932,173.	1,122,921.	
EVAP SW PUMP	509,021.	667,809.	750,878.	
COND SW PUMP	491,487.	598,335.	833,670.	
NH <sub>3</sub> CIRC PUMP	114,361.	135,329.	169,505.	
NH <sub>3</sub> RE-FLUX PUMP	53,845.	63,756.	79,790.	
OPTIMUM COST (\$)	48,852,480.	70,020,000.	104,149,488.	

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COST/KW (NET) OUTPUT  
(\$/KW)

4885.25      4668.00      5207.47



TABLE 3: Heat Exchanger Comparisons (Titanium Tubed)

EVAPORATOR	10 MW	15 MW	20 MW	25 MW
HT ABSORB (BTU/HR)	1.44 E09	2.01 E09	2.80 E09	3.46 E09
SW FLOW (LB <sub>m</sub> /HR)	2.27 E08	3.34 E08	4.14 E08	4.86 E08
SW TEMP IN (DEG F)	80.0	80.0	80.0	80.0
SW TEMP OUT (DEG F)	73.36	73.72	72.92	72.56
NH <sub>3</sub> FLOW (LB <sub>m</sub> /HR)	2.73 E06	3.78 E06	5.29 E06	6.53 E06
OPER PRESS (LB <sub>f</sub> /IN <sup>2</sup> )	129.0	130.1	127.72	127.1
SAT TEMP (DEG F)	70.11	70.59	69.54	69.27
OUTLET TEMP (DEG F)	70.06	70.51	69.47	69.19
OUTLET QUALITY (PCT)	92	92	92	92
NH <sub>3</sub> PRESS DROP (LB <sub>f</sub> /IN <sup>2</sup> )	0.105	0.162	0.165	0.174
TUBE CHARACTERISTICS				
OUTER DIA (IN)	0.947	0.952	0.945	0.929
WALL THICK (IN)	0.025	0.025	0.025	0.025
LENGTH (FT)	43.66	42.18	42.27	42.56

TABLE 3: Heat Exchanger Comparisons (Continued)

EVAPORATOR	10 MW	15 MW	20 MW	25 MW
TUBE PROFILE - STAGGERED EQUILATERAL TRIANGLE				
PITCH RATIO	1.4	1.4	1.47	1.52
ENHANCEMENT - PLAIN TUBE				
SW VEL (FT/SEC)	6.3	6.03	5.80	5.64
T WALL (DEG F)	71.09	71.47	70.51	70.22
FILM TEMP (DEG F)	70.57	70.99	69.99	69.71
DELTA T BOILING (DEG F)	1.027	0.952	1.041	1.032
LMTD	6.00	5.76	6.31	6.34
EFFECTIVENESS	0.671	0.667	0.677	0.693
NTU	1.112	1.098	1.13	1.182
OVLT HT COEF (BTU/ HR·FT <sup>2</sup> ·F)	623.19	612.97	601.47	595.73
h (WATER)	1156.01	1115.97	1081.6	1059.15
h (FOULING)	3787.7	3788.5	3787.35	3784.37

TABLE 3: Heat Exchanger Comparisons (Continued)

EVAPORATOR	10 MW	15 MW	20 MW	25 MW
h (METAL)	4413.77	4404.7	4415.43	4429.66
h (AMMONIA)	4015.73	4088.4	4040.26	4090.74
SW PRESS DROP ( $\text{LB}_f/\text{IN}^2$ )	4.593	4.061	3.824	3.727
MOISTURE SEPARATOR				
OPER PRESS ( $\text{LB}_f/\text{IN}^2$ )	128.9	129.94	127.55	126.92
OUTLET TEMP (DEG F)	69.92	70.33	69.22	68.89
OUTLET QUALITY (PCT)	99.5	99.5	99.5	99.5
$\text{NH}_3$ PRESS DROP ( $\text{LB}_f/\text{IN}^2$ )	0.311	0.416	0.578	0.681
CONDENSER				
HT REJECT (BTU/HR)	1.44 E09	2.01 E09	2.80 E09	3.46 E09
SW FLOW ( $\text{LB}_m/\text{HR}$ )	2.27 E08	3.34 E08	4.14 E08	4.86 E08

TABLE 3: Heat Exchanger Comparisons (Continued)

CONDENSER	10 MW	15 MW	20 MW	25 MW
SW TEMP IN (DEG F)	40.0	40.0	40.0	40.0
SW TEMP OUT (DEG F)	46.29	46.06	46.00	46.16
NH <sub>3</sub> FLOW (LB <sub>m</sub> /HR)	2.73 E06	3.79 E06	5.29 E06	6.53 E06
OPER PRESS (LB <sub>f</sub> /IN <sup>2</sup> )	88.16	88.15	87.46	87.49
SAT TEMP (DEG F)	49.36	49.36	49.95	48.97
OUTLET TEMP (DEG F)	49.3	49.24	49.84	48.85
NH <sub>3</sub> PRESS DROP (LB <sub>f</sub> /IN <sup>2</sup> )	0.127	0.206	0.173	0.195
TUBE CHARACTERISTICS				
OUTER DIA (IN)	0.935	0.972	0.957	0.940
WALL THICK (IN)	0.025	0.025	0.025	0.025
LENGTH (FT)	58.57	57.42	58.32	59.09
TUBE PROFILE - STAGGERED EQUILATERAL TRIANGLE				
PITCH RATIO	1.4	1.4	1.48	1.51

TABLE 3: Heat Exchanger Comparisons (Continued)

CONDENSER	10 MW	15 MW	20 MW	25 MW
ENHANCEMENT - PLAIN TUBE				
SW VEL (FT/SEC)	6.12	6.02	5.72	5.60
T WALL (DEG F)	48.39	48.33	48.04	48.07
FILM TEMP (DEG F)	48.84	48.79	48.44	48.46
DELTA T COND (DEG F)	0.895	0.901	0.808	0.781
LMTD	5.56	5.68	5.29	5.17
EFFECTIVENESS	0.678	0.655	0.679	0.696
NTU	1.133	1.065	1.136	1.191
OVL HT COEF (BTU/ HR·FT <sup>2</sup> ·F)	454.4	446.83	438.4	435.73
h (WATER)	719.6	704.4	678.0	669.3
h (FOULING)	3785.5	3792.0	3789.4	3786.4
h (METAL)	4424.2	4393.6	4405.6	4420.1
h (AMMONIA)	3118.9	3058.6	3171.5	3219.8
SW PRESS DROP (LB <sub>f</sub> /IN <sup>2</sup> )	6.31	5.64	5.30	5.28

TABLE 4: Heat Exchanger Comparisons (Aluminum Tubed)

EVAPORATOR	10 MW	15 MW	20 MW	25 MW
HT ABSORB (BTU/HR)	1.51 E09	1.97 E09	2.80 E09	
SW FLOW (LB <sub>m</sub> /HR)	2.53 E08	3.42 E08	3.89 E08	
SW TEMP IN (DEG F)	80.0	80.0	80.0	
SW TEMP OUT (DEG F)	73.74	73.99	72.47	
NH <sub>3</sub> FLOW (LB <sub>m</sub> /HR)	2.86 E06	3.72 E06	5.29 E06	
OPER PRESS (LB <sub>f</sub> /IN <sup>2</sup> )	129.57	131.98	128.01	
SAT TEMP (DEG F)	70.35	71.4	69.67	
OUTLET TEMP (DEG F)	70.32	71.4	69.61	
OUTLET QUALITY (PCT)	92.0	92.0	92.0	
NH <sub>3</sub> PRESS DROP (LB <sub>f</sub> /IN <sup>2</sup> )	0.070	0.087	0.136	
TUBE CHARACTERISTICS				
OUTER DIA (IN)	1.221	1.046	0.982	
WALL THICK (IN)	0.065	0.065	0.065	

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TABLE 4: Heat Exchanger Comparisons (Continued)

EVAPORATOR	10 MW	15 MW	20 MW	25 MW
LENGTH (FT)	47.68	42.42	41.63	
TUBE PROFILE - STAGGERED EQUILATERAL TRIANGLE				
PITCH RATIO	1.4	1.46	1.50	
ENHANCEMENT - PLAIN TUBE				
SW VEL (FT/SEC)	6.6	6.2	5.89	
T WALL (DEG F)	71.49	72.25	70.63	
FILM TEMP (DEG F)	70.90	71.80	70.12	
DELTA T BOILING (DEG F)	1.163	0.885	1.02	
LMTD	6.0	5.03	5.8	
EFFECTIVENESS	0.649	0.699	0.729	
NTU	1.047	1.199	1.307	
OVL HT COEF (BTU/ HR·FT <sup>2</sup> ·F)	646.7	643.2	625.5	
h (WATER)	1090.0	1052.5	1011.5	

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TABLE 4: Heat Exchanger Comparisons (Continued)

25 MW

EVAPORATOR

20 MW

15 MW

10 MW

$h$  (FOULING)

3470.5

3503.1

3574.5

$h$  (METAL)

13,252.2

13,312.9

13,444.8

$h$  (AMMONIA)

4058.5

4097.4

5040.3

SW PRESS DROP ( $\text{LB}_f/\text{IN}^2$ )

4.11

4.22

4.32

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MOISTURE SEPARATOR

127.88

OPER PRESS ( $\text{LB}_f/\text{IN}^2$ )

131.89

129.5

OUTLET TEMP (DEG F)

69.38

71.23

70.22

OUTLET QUALITY (PCT)

99.5

99.5

99.5

$\text{NH}_3$  PRESS DROP ( $\text{LB}_f/\text{IN}^2$ )

0.522

0.307

0.24

CONDENSER

2.71 E09

HI REJECT (BTU/HR)

1.89 E09

1.47 E09



TABLE 4: Heat Exchanger Comparisons (Continued)

CONDENSER	10 MW	15 MW	20 MW	25 MW
SW FLOW (LB <sub>m</sub> /HR)	2.44 E08	3.04 E08	4.38 E09	
SW TEMP IN (DEG F)	40.0	40.0	40.0	
SW TEMP OUT (DEG F)	46.30	46.53	46.49	
NH <sub>3</sub> FLOW (LB <sub>m</sub> /HR)	2.86 E06	3.72 E06	5.30 E06	
OPER PRESS (LB <sub>f</sub> /IN <sup>2</sup> )	88.52	89.18	87.83	
SAT TEMP (DEG F)	49.52	49.97	49.17	
OUTLET TEMP (DEG F)	49.52	49.90	49.08	
NH <sub>3</sub> PRESS DROP (LB <sub>f</sub> /IN <sup>2</sup> )	0.088	0.111	0.148	
TUBE CHARACTERISTICS				
OUTER DIA (IN)	1.176	1.092	1.001	
WALL THICK (IN)	0.065	0.065	0.065	
LENGTH (FT)	64.04	57.46	57.24	
TUBE PROFILE - STAGGERED EQUILATERAL TRIANGLE				

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TABLE 4: Heat Exchanger Comparisons (Continued)

	10 MW	15 MW	20 MW	25 MW
CONDENSER				
PITCH RATIO	1.4	1.46	1.5	
ENHANCEMENT - PLAIN TUBE				
SW VEL (FT/SEC)	6.31	6.23	5.87	
T WALL (DEG F)	48.51	48.85	48.25	
FILM TEMP (DEG F)	49.02	49.38	48.67	
DELTA T COND (DEG F)	1.020	1.055	0.832	
LMTD	5.820	6.062	5.175	
EFFECTIVENESS	0.661	0.660	0.715	
NTU	1.082	1.078	1.254	
OVL HT COEF (BTU/ HR·FT <sup>2</sup> ·F)	454.57	453.56	446.48	
h (WATER)	670.0	668.56	641.97	
h (FOULING)	3557.0	3523.7	3480.59	
h (METAL)	13,414.54	13,351.18	13,271.11	
h (AMMONIA)	2842.96	2853.65	3131.0	
SW PRESS DROP (LB <sub>f</sub> /IN <sup>2</sup> )	5.92	5.66	5.74	

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# APPENDIX A

## SAMPLE INPUT DATA FOR OTEC ANALYSIS

### EVAPORATOR - HORIZONTAL

TUBE O.D.	1.000(IN)	25.400(MM)
TUBE LENGTH	40.000(FT)	12.192(M)
SW TUBE VEL	6.000(FT/S)	1.829(M/S)
OPER PRESSURE	130.000(LBF/IN2)	0.896(MPA)
TUBE MATERIAL - TITANIUM		
THERMAL COND(K)	9.500(BTU/HR.FT.F)	16.502(W/M.C)
TUBE PROFILE - STAGGERED EQUI-LATERAL		
PITCH RATIO	1.50	
ENHANCEMENT - PLAIN TUBE		

### CONDENSER - HORIZONTAL

TUBE O.D.	1.000(IN)	25.400(MM)
TUBE LENGTH	56.500(FT)	17.221(M)
SW TUBE VEL	6.000(FT/S)	1.829(M/S)
OPER PRESSURE	89.000(LBF/IN2)	0.614(MPA)
TUBE MATERIAL - TITANIUM		
THERMAL COND(K)	9.500(BTU/HR.FT.F)	16.502(W/M.C)
TUBE PROFILE - STAGGERED EQUI-LATERAL		
PITCH RATIO	1.50	
ENHANCEMENT - PLAIN TUBE		

### SALT WATER HOT PIPE

PIPE I.D.	19.300(FT)	5.883(M)
PIPE LENGTH	300.000(FT)	91.440(M)
SW PIPE VEL	4.500(FT/S)	1.372(M/S)
SW INLET TEMP	80.000(DEG F)	26.667(DEG C)
SW SALINITY	35.0/000	

### SALT WATER COLD PIPE

PIPE I.D.	19.600(FT)	5.974(M)
PIPE LENGTH	3000.000(FT)	914.400(M)
SW PIPE VEL	5.500(FT/S)	1.676(M/S)
SW INLET TEMP	40.000(DEG F)	4.444(DEG C)
SW SALINITY	35.0/000	

AMMONIA CIRC PIPE		
PIPE I.D.	2.000(FT)	0.610(M)
PIPE LENGTH	150.000(FT)	45.720(M)
AMMONIA RE-FLUX PIPE		
PIPE I.D.	2.000(FT)	0.610(M)
PIPE LENGTH	50.000(FT)	15.240(M)
PUMP AND GEN-TURB PERFORMANCE		
EVAP SW PUMP		
EFFICIENCY MECH	95.00(PCT)	MOTOR 98.00(PCT)
COND SW PUMP		
EFFICIENCY MECH	85.00(PCT)	MOTOR 98.00(PCT)
AMMONIA CIRC PUMP		
EFFICIENCY MECH	75.00(PCT)	MOTOR 98.00(PCT)
GEN-TURB EFFICIENCIES		
GEN MECH&ELECT	96.60(PCT)	
TURB MECH	99.80(PCT)	
POWER REQUIREMENTS		
NET POWER OUTPUT	15.000(MW)	

# APPENDIX B

## SAMPLE OTEC ANALYSIS OPTIMIZATION OUTPUT DATA

### EVAPORATOR - HORIZONTAL

HT ABSORB	2005013760.0(BTU/HR)	537.601(MW)
SW FLOW	334081230.0(LBM/HR)	151535904.0(KG/HR)
SW TEMP IN	80.000(DEG F)	26.667(DEG C)
SW TEMP OUT	73.724(DEG F)	23.130(DEG C)
NH3 FLOW	3788703.0(LBM/HR)	1718519.0(KG/HR)
OPER PRESSURE	130.097(LBF/IN2)	396.987(KPA)
EVAP SAT TEMP	70.535(DEG F)	21.436(DEG C)
OUTLET TEMP	70.514(DEG F)	21.397(DEG C)
OUTLET QUALITY	92.00(PCT)	
NH3 PRESS DROP	0.162(LBF/IN2)	1.115(KPA)

### TUBE CHARACTERISTICS

OUTER DIA	0.952(IN)	24.180(MM)
WALL THICK	0.025(IN)	0.639(MM)
LENGTH	42.132(FT)	12.857(M)

### MATERIAL - TITANIUM

### TUBE PROFILE - STAGGERED EQUI-LATERAL

PITCH RATIO 1.40

### ENHANCEMENT - PLAIN TUBE

SW VELOCITY	6.026(FT/S)	1.837(M/S)
T WALL(SHELLSIDE)	71.466(DEG F)	21.926(DEG C)
FILM TEMP	70.990(DEG F)	21.661(DEG C)
DELTA T BOILING	0.952(DEG F)	0.529(DEG C)
L.M.T.D.	5.757(DEG F)	3.198(DEG C)
EVAP EFFECTIVENESS	0.667	
NR OF TRANSFER UNITS	1.093	
UVL HT COEF	612.97(BTU/HR.FT2.F)	3430.58(W/12.C)
H(WATER)	1115.97(BTU/HR.FT2.F)	6336.71(W/12.C)
H(FOULING)	3738.54(BTU/HR.FT2.F)	21512.09(W/12.C)
H(METAL)	4409.72(BTU/HR.FT2.F)	25039.23(W/12.C)
H(AMMONIA)	4088.44(BTU/HR.FT2.F)	23215.00(W/12.C)

HT SURFACE	572405.00(FT <sup>2</sup> )	53178.12(42)
TUBE SHEET DIA	27.213(FT)	8.294(M)
TOT NR OF TUBES	5449.	
SW PRESS DROP	4.061(LBF/IN <sup>2</sup> )	23.003(KPA)
MOISTURE SEPARATOR-INSIDE EVAP SHELL		
OPER PRESSURE	129.935(LBF/IN <sup>2</sup> )	395.872(KPA)
OUTLET TEMP	70.333(DEG F)	21.296(DEG C)
OUTLET QUALITY	99.50(PCT)	
NH3 PRESS DROP	0.416(LBF/IN <sup>2</sup> )	2.353(KPA)

#### CONDENSER - HORIZONTAL

HT REJECT	1935472896.0(BTU/HR)	567.221(M)
SW FLOW	334871552.0(LBM/HR)	151894363.0(KG/HR)
SW TEMP IN	40.000(DEG F)	4.444(DEG C)
SW TEMP OUT	46.055(DEG F)	7.308(DEG C)
NH3 FLCW	3738708.0(LBM/HR)	1713519.0(KG/HR)
OPER PRESSURE	63.151(LBF/IN <sup>2</sup> )	607.731(KPA)
COND SAT TEMP	49.351(DEG F)	9.645(DEG C)
OUTLET TEMP	49.238(DEG F)	9.577(DEG C)
NH3 PRESS DROP	0.206(LBF/IN <sup>2</sup> )	1.423(KPA)

#### TUBE CHARACTERISTICS

OUTTER DIA	0.972(IN)	24.683(M)
WALL THICK	0.025(IN)	0.642(M)
LENGTH	57.416(FT)	17.500(M)

#### MATERIAL - TITANIUM

#### TUBE PROFILE - STAGGERED EQUI-LATERAL

PITCH RATIO 1.40

#### ENHANCEMENT - PLAIN TUBE

SW VELOCITY	6.017(FT/S)	1.834(M/S)
T WALL(SHELLSIDE)	48.331(DEG F)	9.073(DEG C)
FILM TEMP	48.785(DEG F)	9.325(DEG C)
DELTA T COND	0.907(DEG F)	0.504(DEG C)
L.M.T.D.	5.683(DEG F)	3.157(DEG C)
COND EFFECTIVENESS	0.655	
NR OF TRANSFER UNITS	1.065	

CVL HT COEF	445.33(BTU/HR.FT <sup>2</sup> .F)	2537.13(W/M <sup>2</sup> .C)
H(WATER)	704.35(BTU/HR.FT <sup>2</sup> .F)	3999.47(W/M <sup>2</sup> .C)
H(FOULING)	3792.00(BTU/HR.FT <sup>2</sup> .F)	21531.75(W/M <sup>2</sup> .C)
H(METAL)	4593.63(BTU/HR.FT <sup>2</sup> .F)	24947.90(W/M <sup>2</sup> .C)
H(AMMONIA)	3053.55(BTU/HR.FT <sup>2</sup> .F)	17367.63(W/M <sup>2</sup> .C)
HT SURFACE	762190.25(F <sup>2</sup> )	70809.69(M <sup>2</sup> )
TUBE SHEET DIA	27.194(FT)	9.289(M)
TOT NR OF TUBES	52179.	
SW PRESS DROP	5.636(LBF/IN <sup>2</sup> )	33.361(KPA)

#### SALT WATER HOT PIPE

PIPE I.D.	20.077(FT)	6.120(M)
PIPE LENGTH	300.000(FT)	91.440(M)
SW PIPE VEL	4.537(FT/S)	1.401(M/S)
SW FLOW	334081280.0(LB <sup>3</sup> /HR)	151535904.0(KG/HR)
SW INLET TEMP	80.000(DEG F)	26.667(DEG C)
SW SALINITY	35.0/000	
SW PRESS DROP	0.322(LBF/IN <sup>2</sup> )	2.217(KPA)

#### SALT WATER COLD PIPE

PIPE I.D.	18.622(FT)	5.676(M)
PIPE LENGTH	3000.000(FT)	914.400(M)
SW PIPE VEL	5.334(FT/S)	1.625(M/S)
SW FLOW	334871552.0(LB <sup>3</sup> /HR)	151394368.0(KG/HR)
SW INLET TEMP	40.000(DEG F)	4.444(DEG C)
SW SALINITY	35.0/000	
SW PRESS DROP	0.508(LBF/IN <sup>2</sup> )	3.501(KPA)

#### AMMONIA CIRC PIPE

PIPE I.D.	2.001(FT)	0.610(M)
PIPE LENGTH	150.000(FT)	45.720(M)
NH3 FLOW	3788708.0(LB <sup>3</sup> /HR)	1718519.0(KG/HR)
NH3 PRESS DROP	15.033(LBF/IN <sup>2</sup> )	103.650(KPA)

#### AMMONIA RE-FLUX CIRC PIPE

PIPE I.D.	2.000(FT)	0.610(M)
PIPE LENGTH	50.000(FT)	15.240(M)
NH3 FLOW	1136612.0(LB <sup>3</sup> /HR)	515555.3(KG/HR)
NH3 PRESS DROP	9.874(LBF/IN <sup>2</sup> )	68.079(KPA)

# PUMP AND GEN-TURB PERFORMANCE

## EVAP SW PUMP

HEAD PRESS	9.898(FT)	3.017(M)
CAPACITY	555259.5(GAL/MIN)	2472587.0(LIT/MIN)
EFFICIENCY MECH	35.00(PCT)	MOTOR 98.00(PCT)

## COND SW PUMP

HEAD PRESS	20.753(FT)	9.227(M)
CAPACITY	652119.2(GAL/MIN)	2468271.0(LIT/MIN)
EFFICIENCY MECH	35.00(PCT)	MOTOR 98.00(PCT)

## AMMONIA CIRC PUMP

HEAD PRESS	212.327(FT)	64.717(M)
CAPACITY	12101.7(GAL/MIN)	45305.0(LIT/MIN)
EFFICIENCY MECH	75.00(PCT)	MOTOR 99.00(PCT)

## AMMONIA RE-FLUX PUMP

HEAD PRESS	33.016(FT)	11.597(M)
CAPACITY	5752.4(GAL/MIN)	14127.1(LIT/MIN)
EFFICIENCY MECH	75.00(PCT)	MOTOR 98.00(PCT)

## GEN-TURB EFFICIENCIES

GEN MECH&ELECT	96.60(PCT)
TURB MECH	99.80(PCT)
TURB INTERNAL	99.83(PCT)

TURB OUTLET QUALITY 96.77(PCT)

## POWER REQUIREMENTS

TURB-GEN GROSS	27668.313(HP)	20.633(MW)
EFFICIENCY LOSSES		0.559(MW)
EVAP SW PUMP	1964.851(HP)	1.496(MW)
COND SW PUMP	4129.313(HP)	3.143(MW)
NH3 CIRC PUMP	541.714(HP)	0.412(MW)
NH3 RE-FLUX PUMP	29.097(HP)	0.022(MW)

NET POWER OUTPUT

15.000(MW)

PERCENT PARASITIC POWER 24.59(PCT)

THERMODYNAMIC CYCLE EFFICIENCY 2.65(PCT)



# COST OF COMPONENTS

EVAPORATOR	3223672.00(DOLLARS)
CONDENSER	3567154.00(DOLLARS)
GEN-TURBINE	1578776.00(DOLLARS)
GENERATOR	937205.06(DOLLARS)
EVAP SW PUMP	653332.94(DOLLARS)
COND SW PUMP	552298.06(DOLLARS)
NH3 CIRC PUMP	136824.94(DOLLARS)
NH3 RE-FLUX PUMP	64448.34(DOLLARS)

CPTIMUM COST	----- 20949216.00(DOLLARS)
COST PER NET KW OUTPUT	1389.95(DOLLARS)

# APPENDIX C

## SAMPLE COPEs OPTIMIZATION AND SENSITIVITY ANALYSIS DATA

```

$BLOCK A (TITLE CARD)
OCEAN THERMAL ENERGY CONVERSION (OTEC) POWER SYSTEM
$BLOCK B (PROGRAM CONTROL PARAMETERS)
2,16,16
$BLOCK C (INTEGER OPT CONTROL PARAMETERS)
5,0,0,5
$BLOCK D (FLOATING PT OPT PRG PARAMETERS)
0.0
0.0
0.0
$BLOCK E (TOT NR DESIGN VAR, DESIGN OBJ IDENT AND SIGN)
16,27,-1.0
$BLOCK F (DESIGN VARIABLE BOUNDS, INIT VALUES & SCALE FACTOR)
1.0,1.0+20
1.0,1.0+20
1.0,1.0+20
1.0,1.0+20
1.0,1.0+20
85.0,1+3.0
85.0,1+8.0
0.5,2.5
0.5,2.5
10.0,1.0+20
10.0,1.0+20
2.0,10.0
2.0,10.0
2.0,10.0
2.0,10.0
2.0,10.0
1.4,3.0
1.4,3.0
$BLOCK G (DESIGN VARIABLE IDENT)
1,1,1.0
2,2,1.0
3,3,1.0
4,4,1.0
5,5,1.0

```

6,6,1.0	5	5	1.0
7,7,1.0	7	7	1.0
8,8,1.0	8	8	1.0
9,9,1.0	9	9	1.0
10,10,1.0	10	10	1.0
11,11,1.0	11	11	1.0
12,12,1.0	12	12	1.0
13,13,1.0	13	13	1.0
14,14,1.0	14	14	1.0
15,15,1.0	15	15	1.0
16,16,1.0	16	16	1.0

\$BLOCK H (NR OF CONSTRAINED PARAMETERS)  
5

\$BLOCK I (CONSTRAINT IDENT AND BOUNDS)  
17

1.,0.,0.,3.,0.0	0.0	3.	0.0
18,20	20		
0.1,0.0,1.0+20,0.0	0.0	1.0+20	0.0
21,23	23		
0.,0.0,1.0+20,0.0	0.0	1.0+20	0.0
24			
30.,0.0,90.,0.0	0.0	90.	0.0
25,26	26		
10.,0.0,1.0+20,0.0	0.0	1.0+20	0.0

\$BLOCK P (SENSITIVITY OBJECTIVES)  
27,0

1,2,3,4,5,6,7,8,9,10,11,12,13,14,15,16,17,18,19,20,

1	2	3	4	5
9	10	11	12	13
17	18	19	20	21
25	26	27		
21,22,23,24,25,26,27				
14	15	16		
22	23	24		

\$BLOCK Q (SENSITIVITY VARIABLE BOUNDS)  
1,6

18.,10.,15.,18.,20.,25.	15.	18.	20.	25.
2,6				
18.,10.,15.,18.,20.,25.	15.	18.	20.	25.
3,6				

2., 0.5, 1. <sup>3</sup> , 2., 3., 4. <sup>0</sup>	1.	2.	3.	4.
4, 0 2. 0.5				
2., 0.5, 1. <sup>4</sup> , 2., 3., 4. <sup>6</sup>	1.	2.	3.	4.
5, 6 2. 0.5				
130., 128. <sup>5</sup> , 129., 130. <sup>6</sup> , 131., 132. <sup>6</sup>	130.	131.	132.	
6, 6 130. 129.				
89., 87. <sup>6</sup> , 88., 89., 90. <sup>6</sup> , 91. <sup>6</sup>	89.	90.	91.	
7, 6 89. 87.				
1., .5., .75 <sup>7</sup> , 1., 1.25, 1. <sup>6</sup> .5	.75	1.	1.25	1.5
8, 5 1. .5				
1., .5., .75 <sup>9</sup> , 1., 1.25, 1. <sup>6</sup> .5	.75	1.	1.25	1.5
9, 6 1. .5				
40., 30. <sup>9</sup> , 35., 40., 45. <sup>6</sup> , 50. <sup>6</sup>	35.	40.	45.	50.
10, 5 40. 30.				
55., 45. <sup>10</sup> , 50., 55., 60. <sup>6</sup> , 65. <sup>6</sup>	50.	55.	60.	65.
11, 6 55. 45.				
7., 5., 6. <sup>11</sup> , 7., 8., 9. <sup>6</sup>	5.	6.	7.	8.
12, 6 7. 5.				
4., 2., 3. <sup>12</sup> , 4., 5., 6. <sup>6</sup>	2.	3.	4.	5.
13, 6 4. 2.				
7., 5., 6. <sup>13</sup> , 7., 8., 9. <sup>6</sup>	5.	6.	7.	8.
14, 6 7. 5.				
4., 2., 3. <sup>14</sup> , 4., 5., 6. <sup>6</sup>	2.	3.	4.	5.
15, 6 4. 2.				
1.0, 1.4, 1. <sup>15</sup> .5, 1.0, 1.7, 1.8 <sup>6</sup>	1.5	1.0	1.7	1.8
16, 6 1.6 1.4				
1.0, 1.4, 1. <sup>16</sup> .5, 1.6, 1.7, 1.3 <sup>6</sup>	1.5	1.0	1.7	1.8
16, 6 1.6 1.4				
END				



CHSE - EVAP HEAT EXCHANGER SHELL COST(\$)  
 CHT - VERT. DIST. BETWEEN TUBES-CENTER TO CENTER (IN)  
 CL - COND. TUBE PROFILE CHANNEL LENGTH (FT)  
 CLMID - CONDENSER LOG MEAN TEMPERATURE DIFFERENCE (DEG F)  
 CLMIDC - CONVERSION OF CLMID TO S.I. UNITS (DEG C)  
 CMINC - CAPACITY RATE OF CONDENSER SW (BTU/HR.F)  
 CMINE - EVAP SW CAPACITY RATE (BTU/HR.F)  
 CMOM - TWO-PHASE EFFECT OF COND DROP MOMENTUM (LBF/IN2)  
 CMW - 746 MW/HP  
 CNH3P - COST OF AMMONIA CIRC PUMP (DOLLARS)  
 CNH3RP - COST OF AMMONIA RE-FLUX PUMP (\$)  
 CNTU - NUMBER OF CONDENSER TRANSFER UNITS  
 CONC - COLD PIPE SALINITY CONCENTRATION  
 CONH - HOT PIPE SALINITY CONCENTRATION  
 CORRF - CORRECTION FACTOR FOR REVISED NH3 HT COEF  
 CPKE - CONDENSER OUTLET RESISTANCE COEFFICIENT  
 CPKI - CONDENSER INLET RESISTANCE COEFFICIENT  
 CPLAT - VERT. DIST. BETWEEN TUBES-CENTER TO CENTER (IN)  
 CPLONG - HORIZ. DIST. BETWEEN TUBES-CENTER TO CENTER (IN)  
 CPNH3E - SPECIFIC HEAT OF AMMONIA IN EVAP (BTU/LBM.F)  
 CPNH3 (EFT) - SUBROUTINE SPECIFIC HEAT PITCH RATIO (IN)  
 CPR - CONDENSER TUBE PROFILE PITCH RATIO (IN)  
 CPSWC - EVAP SW SPECIFIC HEAT (BTU/LBM.F)  
 CPSW (TBLKE) - SUBROUTINE SPECIFIC HEAT (BTU/LBM.F)  
 CPSW (TBLKE) - SUBROUTINE SW SPECIFIC HEAT (BTU/LBM.F)  
 CRKE - CONDENSER OUTLET RESISTANCE COEFFICIENT  
 CRKI - CONDENSER INLET RESISTANCE COEFFICIENT  
 CSF - FLUID-SURFACE COMBINATION VALUE  
 CTAREA - CONDENSER TUBE PROFILE AREA-CENTER TO CENTER (IN2)  
 CTIC - CONDENSER TUBE INSTALLATION COST (\$)  
 CTIE - EVAP TUBE INSTALLATION COST (\$)  
 CTMC - CONDENSER TUBE MATERIAL COST (\$)  
 CTME - EVAP TUBE MATERIAL COST (\$)  
 CTRIC - THERMAL RESISTANCE FOR COND. TUBE SW (HR.F/BTU)  
 CTR2C - THERMAL RESISTANCE FOR COND. SW FOULING (HR.F/BTU)  
 CTR3C - THERMAL RESISTANCE FOR WALL THICK (HR.F/BTU)  
 CTR5C - THERMAL RESISTANCE FOR AMMONIA (HR.F/BTU)  
 CTRIE - THERMAL RESIST EVAP (TUBE) SW (HR.F/BTU)  
 CTR2E - THERMAL RESIST EVAP (TUBE) SW FOULING (HR.F/BTU)  
 CTR3E - THERMAL RESIST EVAP (TUBE) WALL THICK (HR.F/BTU)  
 CTR5E - THERMAL RESIST EVAP (TUBE) FOR AMMONIA (HR.F/BTU)  
 CTSLE - CONDENSER TUBE SHEET LABOR COST (\$)  
 CTSLE - EVAP TUBE SHEET LABOR (\$)  
 CTSME - CONDENSER TUBE SHEET MATERIAL COST (\$)  
 CTSME - EVAP TUBE SHEET MATERIAL COST (\$)  
 CTURB - COST OF AMMONIA TURBINE (\$)  
 CTWL - CONDENSER TUBE SIDE WALL TEMPERATURE (DEG F)

NPU05400  
 NPU01050  
 NPU00840  
 NPU01150  
 NPU02530  
 NPU00700  
 NPU04670  
 NPU01820  
 NPU03400  
 NPU06720  
 NPU00270  
 NPU01210  
 NPU03380  
 NPU03390  
 NPU02060  
 NPU06260  
 NPU06250  
 NPU01110  
 NPU01100  
 NPU05110  
 NPU03660  
 NPU00680  
 NPU04650  
 NPU00810  
 NPU04660  
 NPU06360  
 NPU06350  
 NPU05280  
 NPU01090  
 NPU02290  
 NPU05390  
 NPU02280  
 NPU05380  
 NPU01390  
 NPU01410  
 NPU01960  
 NPU01430  
 NPU04940  
 NPU04950  
 NPU04960  
 NPU04970  
 NPU02260  
 NPU05360  
 NPU02270  
 NPU05370  
 NPU00610  
 NPU01450

C	CTW2	-	CONDENSER SHELLSIDE WALL TEMPERATURE( DEG F)	NPU01460
C	CTW2C	-	CONVERSION OF CTW2 TO S.I. UNITS( DEG C)	NPU02550
C	CTWC	-	CONDENSER TUBE WELDING COST(\$)	NPU02370
C	CTWE	-	EVAP TUBE WELDING COST(\$)	NPU05450
C	CWINSE	-	EVAP WATER INLET, NOZZLES, AND SUPPORTS COST(\$)	NPU05440
C	CWINSC	-	COND WATER INLETS, NOZZLES & SUPPORTS COST(\$)	NPU02360
C	DBLKTC	-	ABSOLUTE DIFFERENCE BETWEEN TBLKE & RTBLKE( DEG F)	NPU04900
C	DBLKTC	-	ABSOLUTE DIFFERENCE BETWEEN REVISED AND ASSUMED TBLK TEMP	NPU01170
C	DCONDC	-	TOTAL PRESSURE LOSSES ACROSS CONDENSER( LBF/IN2)	NPU06430
C	DCONDC	-	CONVERSION OF DCONDC TO S.I. UNITS( KPA)	NPU02560
C	DCOREC	-	CONDENSER TUBING FRICTION LOSSES( LBF/IN2)	NPU06420
C	DCOREE	-	EVAPORATOR TUBING FRICTION LOSSES( LBF/IN2)	NPU06080
C	DDENEC	-	PRESSURE LOSSES DUE TO SW DENSITY LOSS( LBF/IN2)	NPU06310
C	DDENEC	-	PRESSURE LOSSES DUE TO SW DENSITY COMPRESSIBILITY( LBF/IN2)	NPU00990
C	DDUCTC	-	LOSSES DUE TO SW DENSITY COMPRESSIBILITY( LBF/IN2)	NPU06270
C	DDUCTH	-	SW COLD PIPE INLET/OUTLET DUCT LOSSES( LBF/IN2)	NPU03110
C	DELB	-	RATIO OF DBLKTC TO SCALB	NPU04920
C	DELB	-	RATIO OF DBLKTC WRT SCALB	NPU01190
C	DELC	-	RATIO OF DHNH3C WRT SCALC	NPU02090
C	DELH	-	RATIO OF HNH3C WITH RESPECT TO HNH3E	NPU05300
C	DELH3	-	RATIO OF DTEMP3 WITH RESPECT TO SCALY	NPU06000
C	DELH4	-	RATIO OF DTEMP4 AND SCALH4( DEG F)	NPU01880
C	DELTAEC	-	CONDENSING TEMPERATURE( DEG F)	NPU01480
C	DELTAEC	-	EVAP DELTA T FOR BOILING( DEG F)	NPU05020
C	DELTAEC	-	CONVERSION OF DELTAC TO S.I. UNITS( DEG C)	NPU02570
C	DELTAEC	-	CONVERSION OF DELTAE TO S.I. UNITS( DEG C)	NPU02580
C	DELTAEC	-	CONVERSION OF DELTAE TO S.I. UNITS( DEG C)	NPU05960
C	DELTAEC	-	RATIO OF DVNH3E WITH RESPECT TO SCALV	NPU06090
C	DEVAPC	-	TOTAL PRESSURE LOSSES ACROSS EVAPORATOR( LBF/IN2)	NPU02590
C	DEVAPC	-	CONVERSION OF DEVAP TO S.I. UNITS( KPA)	NPU00830
C	DF	-	GEN- TURB DOUBLE FLOW FACTOR	NPU00430
C	DH5	-	CONSTRAINT DIFFER FOR ENTHALPY AT STATE PT 5	NPU00980
C	DHEAD	-	LOSSES DUE TO SW DENSITY COMPRESSIBILITY( FT)	NPU02070
C	DHNH3C	-	ABSOLUTE DIFFERENCE BETWEEN HNH3CR AND HNH3C	NPU05290
C	DHNH3C	-	ABSOLUTE DIFFERENCE BETWEEN HNH3E AND HNH3E( BTU/HR.FT2.F)	NPU03490
C	DICPC	-	COLD PIPE INNER DIAMETER( FT)	NPU02600
C	DICPC	-	CONVERSION OF DICP TO S.I. UNITS( M)	NPU03500
C	DIHPC	-	HOT PIPE INNER DIAMETER( FT)	NPU02610
C	DIHPC	-	CONVERSION OF DIHP TO S.I. DIAMETER( FT)	NPU03510
C	DINH3C	-	AMMONIA CIRC PIPE INNER DIAMETER( FT)	NPU02620
C	DINH3C	-	CONVERSION OF DINH3 TO S.I. UNITS( M)	NPU03520
C	DINH3C	-	AMMONIA RE-FLUX PIPE INNER DIAMETER( FT)	NPU02630
C	DINH3C	-	CONVERSION OF DINH3 TO S.I. UNITS( M)	NPU06590
C	DPDME	-	EVAPORATOR SHELLSIDE PRESSURE LOSSES( LBF/IN2)	NPU03870
C	DPDME	-	DEMISTER PRESSURE DROP MULTIPLE	NPU00950
C	DPELEC	-	COND CORE PRESSURE DROP DUE TO RE-FLUX PIPING E2R( LBF/IN2)	NPU00130
C	DPELEC	-	PRESSURE LOSS DUE TO RE-FLUX PIPING E22( LBF/IN2)	NPU06620
C	DPELEC	-	PRESSURE LOSS DUE TO AMMONIA PIPING E22( LBF/IN2)	NPU06630
C	DPELEC	-	TOTAL SW COLD PIPE FRICTION LOSSES( LBF/IN2)	

C	DPIPC	-	CONVERSION OF DPIPC TO S.I. UNITS(KPA)	NPU02640
C	DPIPEC	-	SW COLD PIPE FRICTION LOSSES(LBF/IN2)	NPU06290
C	DPIPEH	-	SW HOT PIPE FRICTION LOSSES(LBF/IN2)	NPU03360
C	DPIPH	-	TOTAL SW HOT PIPE FRICTION LOSSES(LBF/IN2)	NPU04680
C	DPIPHC	-	CONVERSION OF DPIPH TO S.I. UNITS(KPA)	NPU02650
C	DPIPN	-	AMMONIA PCIRC PIPE PRESSURE LOSSES(LBF/IN2)	NPU06630
C	DPIPNR	-	CONVERSION OF DPIPN TO S.I. UNITS(KPA)	NPU02660
C	DPIPNR	-	RE-FLUX AMMONIA PIPE LOSSES(LBF/IN2)	NPU00140
C	DPMCC	-	CONVERSION OF DPMCC TO S.I. UNITS(M)	NPU02680
C	DPMCC	-	CONVERSION OF DPMCC TO S.I. UNITS(M)	NPU02690
C	DPMINE	-	MINOR INLET/OUTLET CONDENSER LOSSES(LBF/IN2)	NPU06370
C	DPMNCC	-	CONVERSION OF DPMNCC TO S.I. UNITS(M)	NPU05620
C	DPMNCC	-	CONVERSION OF DPMNCC TO S.I. UNITS(M)	NPU02670
C	DPMNCC	-	COND. OR SW COLD PIPE PUMP PRESSURE HEAD(LBF/IN2)	NPU06440
C	DPMNCC	-	COND. OR SW COLD PIPE PUMP PRESSURE HEAD(LBF/IN2)	NPU06450
C	DPMPE	-	EVAP. OR SW HOT PIPE PUMP PRESSURE HEAD(FT)	NPU06110
C	DPMPE	-	EVAP. OR SW HOT PIPE PUMP PRESSURE HEAD(LBF/IN2)	NPU06120
C	DPMN	-	AMMONIA CIRC PUMP PRESSURE HEAD(FT)	NPU06650
C	DPMN	-	AMMONIA CIRC PUMP PRESSURE HEAD(LBF/IN2)	NPU06660
C	DPMN	-	RE-FLUX AMMONIA PUMP HEAD(LBF/IN2)	NPU00160
C	DPNH3P	-	AMMONIA CIRCULATION PIPE FRICTION LOSSES(LBF/IN2)	NPU06580
C	DPNHRP	-	RE-FLUX AMMONIA PIPE FRICTION LOSSES(LBF/IN2)	NPU00110
C	DPNRC	-	CONVERSION OF DPINR TO S.I. UNITS(KPA)	NPU02700
C	DPNRC	-	CONVERSION OF DPINR TO S.I. UNITS(M)	NPU02710
C	DPNRC	-	RE-FLUX AMMONIA PUMP HEAD(FT)	NPU00170
C	DPNRC	-	RE-FLUX AMMONIA THERMO. PRESSURE HEAD(LBF/IN2)	NPU06640
C	DPNRC	-	PUMPED RE-FLUX AMMONIA THERMO. PHEAD(LBF/IN2)	NPU00150
C	DPNRC	-	MOISTURE SEPARATOR PRESSURE DROP(LBF/IN2)	NPU05730
C	DSDEM	-	CONVERSION OF DSDEM TO S.I. UNITS(KPA)	NPU02720
C	DSDEM	-	CONVERSION OF DSDEM TO S.I. UNITS(KPA)	NPU05710
C	DSDEM	-	EVAPORATOR SHELLSIDE PRESSURE DROP(LBF/IN2)	NPU02730
C	DSDEM	-	CONVERSION OF DSDEM TO S.I. UNITS(KPA)	NPU02230
C	DSDEM	-	DRILLING TIME/TUBE SHEET THICK COND(MIN/IN)	NPU05340
C	DSDEM	-	EVAP DRILLING TIME/TUBE SHEET THICK COND(MIN/IN)	NPU04370
C	DSDEM	-	CONSTRAINT FOR SAT UPPER TEMP SYSTEM BOUND	NPU04380
C	DSDEM	-	CONSTRAINT FOR SAT LOWER TEMP SYSTEM BOUND	NPU05980
C	DSDEM	-	ABSOLUTE DIFF BETWEEN TIR AND T1(DEG F)	NPU01860
C	DSDEM	-	CONDENSER OUTLET TEMPERATURE BOUND(DEG F)	
C	DSDEM	-	CONSTRAINT FOR SAT CONDENSER TUBE LENGTH	
C	DSDEM	-	CONSTRAINT DIFF BETWEEN VNH3E AND VNH3E(FT/SEC)	NPU02220
C	DSDEM	-	ABSOLUTE TUBING MACHINING COST(\$)	NPU05940
C	DSDEM	-	EVAPORATOR TUBE FRONTAL AREA(FT2)	NPU04050
C	DSDEM	-	EVAPORATOR TUBE FRONTAL AREA(FT2)	NPU04060
C	DSDEM	-	EVAPORATOR TUBE FRONTAL AREA(FT2)	NPU01550
C	DSDEM	-	EVAPORATOR TUBE FRONTAL AREA(FT2)	NPU04620
C	DSDEM	-	EVAPORATOR TUBE FRONTAL AREA(FT2)	NPU04560
C	DSDEM	-	EVAPORATOR TUBE FRONTAL AREA(FT2)	NPU03720
C	DSDEM	-	EVAPORATOR TUBE FRONTAL AREA(FT2)	NPU03750



ECP - COLD PIPE ABSOLUTE ROUGHNESS (FT)  
 EDC - COND SHELLSIDE FLOW CHANNEL DIAMETER (FT)  
 EDE - SHELLSIDE NH3 FLOW CHANNEL DIAMETER (FT)  
 EELEV - TWO-PHASE EFFECT OF EVAP ELEVATION (LBF/IN2)  
 EEP - GENERATOR MECHANICAL & ELECTRICAL EFFICIENCY (PCT)  
 EEP - CONVERSION OF EEP (DECIMAL)  
 EFF - EVAP SHELLSIDE FRICTION FACTOR  
 EFFIC - CONDENSER FIN EFFICIENCY - INSIDE (DECIMAL)  
 EFFIE - EVAPORATOR TUBE FIN EFFICIENCY (DECIMAL)  
 EFFOC - CONDENSER FIN EFFICIENCY - OUTSIDE (DECIMAL)  
 EFFOE - EVAP OUTER TUBE FIN EFFICIENCY (DECIMAL)  
 EFFRICT - TWO-PHASE EFFECT OF EVAP PIPE FRICTION (LBF/IN2)  
 EFFT - EVAP FILM TEMPERATURE (DEG F)  
 EFFTC - CONVERSION OF EFFT TO S.I. UNITS (DEG C)  
 EHP - ABSOLUTE ROUGHNESS FACTOR OF SW HOT PIPE  
 EHT - HALF VERT HEIGHT BETWEEN TUBE, CENTER TO CENTER (IN)  
 ELBQM - 90 DEG ELBOW EQUIV. PIPE DIA (L/D)  
 ELECT - NET ELECTRICAL OUTPUT REQUIRED (MW)  
 ELMTD - EVAP LOG MEAN TEMPERATURE DIFFERENCE (DEG F)  
 ELMTDC - CONVERSION OF ELMTD TO S.I. UNITS (DEG C)  
 EMC - COLD PIPE SW PUMP MOTOR EFFICIENCY (PCT)  
 EME - CONVERSION OF EMC (DECIMAL)  
 EMEC - HOT PIPE SW PUMP MOTOR EFFICIENCY (PCT)  
 EMEH - CONVERSION OF EME (DECIMAL)  
 EMNH3 - AMMONIA CIRC PUMP MOTOR EFFICIENCY (PCT)  
 EMNH3C - CONVERSION OF EMNH3 (DECIMAL)  
 EMNHR - AMMONIA RE-FLUX PUMP MOTOR EFFICIENCY (PCT)  
 EMNHR - CONVERSION OF EMNHR (DECIMAL)  
 EMOM - TWO-PHASE EFFECT OF EVAP DROP MOMENTUM (LBF/IN2)  
 ENH3P - AMMONIA PIPE ABSOLUTE ROUGHNESS (FT)  
 ENTU - NUMBER OF TRANSFER UNITS FOR EVAP  
 EPC - COLD PIPE SW PUMP MECHANICAL EFFICIENCY (PCT)  
 EPEC - CONVERSION OF EPC (DECIMAL)  
 EPEC - HOT PIPE SW PUMP MECHANICAL EFFICIENCY (PCT)  
 EPEL - CONVERSION OF EPE (DECIMAL)  
 EPLAT - VERTICAL HEIGHT BETWEEN TUBE, CENTER TO CENTER (IN)  
 EPLONG - HORIZ DIST BETWEEN TUBE, CENTER TO CENTER (IN)  
 EPNH3 - AMMONIA CIRC PUMP MECHANICAL EFFICIENCY (PCT)  
 EPNH3C - CONVERSION OF EPNH3 (DECIMAL)  
 EPNHR - AMMONIA RE-FLUX PUMP MECHANICAL EFFICIENCY (PCT)  
 EPNHR - CONVERSION OF EPNHR (DECIMAL)  
 EPR - EVAPORATOR TUBE PROFILE PITCH RATIO (IN)  
 EPR - EVAPORATOR EFFECTIVENESS (DECIMAL)  
 ESPACE - EVAP VAPOR FLOW AREA TO MOISTURE SEPARATOR (FT2)  
 ETAREA - TUBE PROFILE AREA PER TUBE (IN2)  
 ETRP - GENERATOR TURBINE MECHANICAL EFFICIENCY (PCT)  
 ETRPC - CONVERSION OF ETRP (DECIMAL)  
 ETURB - INTERNAL TURBINE EFFICIENCY (DECIMAL)

NPU03740  
 NPU01760  
 NPU01570  
 NPU01630  
 NPU03880  
 NPU00320  
 NPU05640  
 NPU01370  
 NPU04750  
 NPU01380  
 NPU04790  
 NPU01600  
 NPU05010  
 NPU02740  
 NPU05690  
 NPU04540  
 NPU03760  
 NPU03980  
 NPU05040  
 NPU02750  
 NPU03890  
 NPU06480  
 NPU03900  
 NPU06150  
 NPU03910  
 NPU06700  
 NPU03920  
 NPU00220  
 NPU01620  
 NPU03770  
 NPU04860  
 NPU03930  
 NPU06460  
 NPU03940  
 NPU06130  
 NPU04610  
 NPU04600  
 NPU03950  
 NPU06670  
 NPU03960  
 NPU00190  
 NPU03650  
 NPU04870  
 NPU04630  
 NPU04570  
 NPU03970  
 NPU00340  
 NPU00590

NPU00600  
 NPU05000  
 NPU04990  
 NPU02760  
 NPU06600  
 NPU03780  
 NPU00120  
 NPU06400  
 NPU06240  
 NPU05650  
 NPU02350  
 NPU06560  
 NPU02770  
 NPU02780  
 NPU06220  
 NPU02790  
 NPU04450  
 NPU02800  
 NPU05580  
 NPU00090  
 NPU00080  
 NPU03410  
 NPU03430  
 NPU03420  
 NPU05670  
 NPU05520  
 NPU05540  
 NPU05810  
 NPU05550  
 NPU05770  
 NPU05800  
 NPU05920  
 NPU05880  
 NPU05910  
 NPU00400  
 NPU05840  
 NPU00440  
 NPU05780  
 NPU05890  
 NPU05530  
 NPU02170  
 NPU02820  
 NPU04820  
 NPU00410  
 NPU05790  
 NPU05900

ETURBP - CONVERSION OF ETURB(PCT)  
 ETW1 - EVAP TUBESIDE WALL TEMPERATURE(DEG F)  
 ETW2 - EVAP SHELLSIDE WALL TEMPERATURE(DEG F)  
 ETW2C - CONVERSION OF ETW2 TO S.I. UNITS(DEG C)  
 EZ1 - DATUM OF AMMONIA PIPING CONFIGURATION  
 EZ2 - ELEVATION OF AMMONIA PIPING(FT)  
 EZR - RE-FLOW AMMONIA PIPING CONFIGURATION HEIGHT(FT)  
 FFCP - FRICTION FACTOR OF SW COLD PIPE  
 FFE - FRICTION FACTOR OF EVAPORATOR PIPING  
 FFHP - FRICTION FACTOR OF SW HOT PIPE  
 FFN43P - FRICTION FACTOR OF PIPED AMMONIA  
 FLN43C - CONVERSION OF FLN43 TO S.I. UNITS(KG/HR)  
 FLN43C - CONVERSION OF FLN43 TO S.I. UNITS(KG/HR)  
 FLN43C - COLD PIPE SW FLOW RATE(LBM/HR)  
 FLOCPC - CONVERSION OF FLOCPC TO S.I. UNITS(KG/HR)  
 FLOHPC - HOT PIPE SW MASS FLOW RATE(LBM/HR)  
 FLOHPC - CONVERSION OF FLOHPC TO S.I. UNITS(KG/HR)  
 FLN43C - AMMONIA MASS FLOW RATE(LBM/HR)  
 FLN43C - RE-FLOW AMMONIA FLOW RATE(LBM/HR)  
 FPF - FRICTION FACTOR OF PIPED RE-FLUX AMMONIA  
 G - PRICING FACTOR FOR DOUBLE-FLOW GEN-TJRBINE  
 GAL - 32.174 FT/SEC2  
 GC - 7.481 GAL/FT3  
 GF - 32.174 FT LBM/LBF·SEC2  
 H1 - NH3 MASS VEL THRU EVAP FREE-FLOW AREA(LBM/FT2·SEC)  
 H2 - ENTHALPY AT STATE PT 2(BTU/LBM)  
 H3 - ENTHALPY OF AMMONIA AT STATE PT 3(BTU/LBM)  
 H3A - ENTHALPY OF SAT LIQ AMMONIA AT STATE PT 3(BTU/LBM)  
 H3F - ENTHALPY OF SAT VAPOR AMMONIA AT STATE PT 3(BTU/LBM)  
 H4 - ENTHALPY OF AMMONIA AT STATE PT 4(BTU/LBM)  
 H4F - ENTHALPY OF SAT LIQ AMMONIA AT STATE PT 4(BTU/LBM)  
 H4G - ENTHALPY OF SAT VAPOR AMMONIA AT STATE PT 4(BTU/LBM)  
 H5 - ENTHALPY AT STATE PT 5(BTU/LBM)  
 H5A - ENTHALPY OF SAT LIQ AMMONIA AT STATE PT 5(BTU/LBM)  
 H5F - ENTHALPY OF SAT VAPOR AMMONIA AT STATE PT 5(BTU/LBM)  
 HF(P3) - SUBROUTINE FOR SAT LIQ AMMONIA(BTU/LBM)  
 HF(P4) - SUBROUTINE FOR SAT LIQ AMMONIA(BTU/LBM)  
 HF(PCOND) - SUBROUTINE FOR ENTHALPY(BTU/LBM)  
 HF(SWC) - PSEUDO HT COEFF COND SW FOULING(BTU/HR·FT2·F)  
 HF(SWC) - CONVERSION OF HF(SWC) TO S.I. UNITS(W/M2·C)  
 HF(SWE) - PSEUDO HT COEFF FOR EVAP SW FOULING(BTU/HR·FT2·F)  
 HF(SWE) - CONVERSION OF HF(SWE) TO S.I. UNITS(W/M2·C)  
 HG5 - ENTHALPY OF VAPOR AT STATE PT 5(BTU/LBM)  
 HG(P3) - SUBROUTINE FOR SAT VAPOR AMMONIA(BTU/LBM)  
 HG(P4) - SUBROUTINE FOR SAT VAPOR AMMONIA(BTU/LBM)

CC HG(PCOND) - SUBROUTINE FOR ENTHALPY OF A VAPOR (BTU/LBM)  
 CC HG(PEVAP) - SUBROUTINE FOR ENTHALPY (BTU/LBM)  
 CC HNH3C - PSEUDO HT COEFF COND AMMONIA (BTU/HR.FT2.F)  
 CC HNH3CC - CONVERSION OF HNH3C TO S.I. UNITS (W/M2.C)  
 CC HNH3EC - PSEUDO HT COEFF FOR EVAP AMMONIA (BTU/HR.FT2.F)  
 CC HNH3EC - CONVERSION OF HNH3E TO S.I. UNITS (W/M2.C)  
 CC HNH3CR - REVISED AMMONIA HT COEFF COND. (BTU/HR.FT2.F)  
 CC HNH3ER - REVISED EVAP SHELLSIDE HT COEFF (BTU/HR.FT2.F)  
 CC HPC - 550 FT.LBF/SEC.HP  
 CC HPKI - INLET DUCT TO SW HOT PIPE RESISTANCE COEFFICIENT  
 CC HPKE - OUTLET SW HOT PIPE RESISTANCE COEFFICIENT  
 CC HSMC - PSEUDO HT COEFF COND SW (BTU/HR.FT2.F)  
 CC HSMCC - CONVERSION OF HSMC TO S.I. UNITS (W/M2.C)  
 CC HSMEC - PSEUDO HT COEFF FOR EVAP SW (BTU/HR.FT2.F)  
 CC HSMEC - CONVERSION OF HSMC TO S.I. UNITS (W/M2.C)  
 CC HTDR - EVAP NH3 HT COEFF DEVELOPING REGION (BTU/HR.FT2.F)  
 CC HTFDR - EVAP NH3 HT COEFF FULLY DEVELOPED REG (BTU/HR.FT2.F)  
 CC HTFSWC - HT COEFF FOR COND. SW FOULING (BTU/HR.FT2.F)  
 CC HTFSWE - EVAP SW FOULING COEFF (BTU/HR.FT2.F)  
 CC HTNH3C - INITIAL AMMONIA HT COEFFICIENT (BTU/HR.FT2.F)  
 CC HTNH3E - EVAP SHELLSIDE HT COEFFICIENT (BTU/HR.FT2.F)  
 CC HTSWC - HT COEFFICIENT FOR CONDENSER SW (BTU/HR.FT2.F)  
 CC HTSWE - EVAP TUBESIDE HT COEFFICIENT (BTU/HR.FT2.F)  
 CC HWC - PSEUDO HT COEFF COND WALL THICK. (BTU/HR.FT2.F)  
 CC HWC - CONVERSION OF HWC TO S.I. UNITS (W/M2.C)  
 CC HWEC - PSEUDO HT COEFF FOR EVAP WALL THICK (BTU/HR.FT2.F)  
 CC HWEC - CONVERSION OF HWC TO S.I. UNITS (W/M2.C)  
 CC MATL - TUBING MATERIAL SELECTION  
 CC OBJ - OBJECTIVE FUNCTION TO MINIMIZE COST  
 CC P1 - PRESSURE OF SAT AMMONIA AT STATE PT 1 (LBF/IN2)  
 CC P3 - PRESSURE OF AMMONIA AT STATE PT 3 (LBF/IN2)  
 CC P3C - CONVERSION OF P3 TO S.I. UNITS (KPA)  
 CC P4 - PRESSURE OF AMMONIA AT STATE PT 4 (LBF/IN2)  
 CC PARAL - PARASITIC PUMP REQUIREMENTS (MW)  
 CC PAVGD - AVG PRESSURE ACROSS MOISTURE SEPARATOR (LBF/IN2)  
 CC PAVGE - AVERAGE EVAPORATOR PRESSURE (LBF/IN2)  
 CC PCOND - OPERATION PRESSURE OF THE CONDENSER (LBF/IN2)  
 CC PCONDC - CONVERSION OF PCOND TO S.I. UNITS (KPA)  
 CC PEVAP - OPERATING PRESSURE OF THE EVAPORATOR (LBF/IN2)  
 CC PEVAPC - CONVERSION OF PEVAP TO S.I. UNITS (KPA)  
 CC PIE - 3.141592654  
 CC PNH3E - PRANDTL NUMBER FOR EVAP SHELLSIDE FLOW  
 CC PNSWC - PRANDTL NUMBER FOR CONDENSER SW  
 CC PNSWE - PRANDTL NUMBER FOR EVAP TUBESIDE FLOW  
 CC PPP - PERCENT OF PARASITIC POWER (PCT)  
 CC PROF - TUBE PROFILE SELECTION  
 CC PRSYST - SYSTEM PRESS RATIO  
 CC PWRCP - POWER COND. OR CSW COLD PIPE PUMP (HP)

NPU00420  
 NPU005570  
 NPU002190  
 NPU002840  
 NPU004840  
 NPU002850  
 NPU002040  
 NPU005160  
 NPU003440  
 NPU002810  
 NPU003080  
 NPU002160  
 NPU002860  
 NPU004810  
 NPU002870  
 NPU005240  
 NPU005250  
 NPU001400  
 NPU004770  
 NPU001250  
 NPU004420  
 NPU001360  
 NPU004740  
 NPU002180  
 NPU002880  
 NPU004830  
 NPU002910  
 NPU003830  
 NPU002490  
 NPU001840  
 NPU005760  
 NPU002890  
 NPU005740  
 NPU006730  
 NPU005830  
 NPU000010  
 NPU003530  
 NPU002900  
 NPU003540  
 NPU002920  
 NPU003450  
 NPU005170  
 NPU001350  
 NPU004730  
 NPU002420  
 NPU003710  
 NPU004360  
 NPU006470

C	PWRCPM	- CONVERSION OF PWRCP(MW)	NPU06490
C	PWREP	- POWER EVAP SW PUMP(HP)	NPU06140
C	PWREP	- CONVERSION OF PWREP(MW)	NPU06160
C	PWRNP	- POWER OF AMMONIA CIRC PUMP(HP)	NPU06680
C	PWRNPC	- CONVERSION OF PWRNP(MW)	NPU00230
C	PWRNP	- POWER AMMONIA RE-FLUX PUMP(HP)	NPU00200
C	PWRNP	- POWER AMMONIA RE-FLUX PUMP(HP)	NPU00860
C	PWRTR	- GEV-TURBINE OUTLET POWER(HP)	NPU00670
C	QC	- AMOUNT OF HEAT REJECTION(BTU/LBM)	NPU02930
C	QC	- CONVERSION OF QC TO S.I. UNITS(MW)	NPU06500
C	QCPMP	- DISCHARGE RATE OF COND SW PUMP(FT3/SEC)	NPU02950
C	QCPMCC	- CONVERSION OF QCPMP TO S.I. UNITS(LIT/MIN)	NPU01440
C	QCT	- HEAT TRANSFERRED PER TUBE(BTU/HR)	NPU05030
C	QEC	- AMOUNT OF EVAPORATOR HEAT ADDITION(BTU/HR)	NPU02940
C	QEPMCC	- CONVERSION OF QEC TO S.I. UNITS(LIT/MIN)	NPU02960
C	QEPMP	- DISCHARGE RATE OF EVAP SW PUMP(FT3/SEC)	NPU06170
C	QIN	- HEAT TRANSFERRED PER EVAP TUBE(BTU/HR)	NPU04980
C	QNP	- AMOUNT OF HEAT ADDED TO THE SYSTEM(MW)	NPU02440
C	QNP	- CONVERSION OF QNP TO S.I. UNITS(LIT/MIN)	NPU02970
C	QNPMP	- DISCHARGE RATE OF AMMONIA CIRC PUMP(FT3/SEC)	NPU06710
C	QRPMP	- CONVERSION OF QRPMP TO S.I. UNITS(LIT/MIN)	NPU02980
C	QRPMP	- DISCHARGE RATE OF AMMONIA RE-FLUX PUMP(FT3/SEC)	NPU00240
C	QRPMP	- DISCHARGE RATE AMMONIA RE-FLUX PUMP(GAL/MIN)	NPU00250
C	RAVG	- AVG SW DENSITY OVER A SPECIFIED DEPTH(LBM/FT3)	NPU00970
C	REMAX	- REYNOLDS NUMBER FOR MAX EVAP SHELLSIDE FLOW	NPU00960
C	RFNH3(CFT)	- SUBROUTINE FOR CONDENSER SHELLSIDE FLOW	NPU05630
C	RFNH3(FT)	- SUBROUTINE FOR AMMONIA DENSITY(LBM/FT3)	NPU01900
C	RFNH3(TI)	- SUBROUTINE FOR AMMONIA DENSITY(LBM/FT3)	NPU01970
C	RGNH3(PEVAP)	- SUBROUTINE SAT LIQ AMMONIA DENSITY(LBM/FT3)	NPU05510
C	RKE	- SW COLD PIPE RESISTANCE COEFFICIENT	NPU05060
C	RKE	- EVAPORATOR OUTLET RESISTANCE COEFFICIENT	NPU05700
C	RKHE	- EVAPORATOR TUBING RESISTANCE COEFFICIENT	NPU06280
C	RKHC	- CONDENSER TUBING RESISTANCE COEFFICIENT	NPU05500
C	RKHP	- SW HOT PIPE RESISTANCE COEFFICIENT	NPU06070
C	RKI	- EVAPORATOR INLET RESISTANCE COEFFICIENT	NPU06410
C	RKNH3P	- PIPED AMMONIA RESISTANCE COEFFICIENT	NPU03330
C	RKNH3C	- PIPED RE-FLUX AMMONIA RESISTANCE COEFFICIENT	NPU05480
C	RHOSWC	- DENSITY OF CONDENSER SW(LBM/FT3)	NPU06570
C	RHOSWE	- EVAPORATOR SW DENSITY(LBM/FT3)	NPU00100
C	RHOSW(TBLC)	- SUBROUTINE SW DENSITY(LBM/FT3)	NPU00730
C	RHOSW(TCIC)	- SUBROUTINE SW DENSITY(LBM/FT3)	NPU00790
C	RHOSW(THIE)	- SUBROUTINE FOR DENSITY OF SW(LBM/FT3)	NPU00740
C	RHOSW(TLCP)	- SUBROUTINE SW DENSITY(LBM/FT3)	NPU00890
C	RHOSWR	- REVISED CONDENSER SW DENSITY(LBM/FT3)	NPU00880
C	RNH3CH	- PSEUDO REYNOLDS NO. FOR HORIZ. CONDENSER	NPU01280
C			NPU02010





TKSWE	-	EVAP	SW	THERMAL	CONDUCTIVITY(BTU/HR.FT.F)	NPU04490
TKSW(TBLKC)	-	SUBROUTINE	FOR	COND.	THERMAL CONDUCT.	NPU01340
TKWC	-	TUBE	MATL	THERMAL	CONDUCTIVITY(BTU/HR.FT.F)	NPU03840
TKWC	-	CONVERSION	OF	TKW	TO S.I. UNITS(W/M.C)	NPU03160
TLCC	-	CONDENSER	TUBE	LENGTH(FT)		NPU03590
TLCC	-	CONVERSION	OF	TLC	TO S.I. UNITS(M)	NPU03170
TLCC	-	COLD	PIPE	LENGTH(FT)		NPU03790
TLCP	-	CONVERSION	OF	TLCP	TO S.I. UNITS(M)	NPU03180
TLCP	-	REVISED	CONDENSER	TUBE	LENGTH(FT)	NPU02210
TLCP	-	CONVERSION	OF	TLCP	TO S.I. UNITS(M)	NPU02990
TLCD	-	TUBE	LENGTH	FOR	DEVELOPING CONVECTIVE FLOW(FT)	NPU05200
TLE	-	EVAPORATOR	TUBE	LENGTH(FT)		NPU03600
TLE	-	CONVERSION	OF	TLE	TO S.I. UNITS(M)	NPU03190
TLE	-	REVISED	EVAP	TUBE	LENGTH(FT)	NPU05320
TLE	-	CONVERSION	OF	TLE	TO S.I. UNITS(M)	NPU03000
TLE	-	TUBE	LENGTH	FOR	FULLY DEVELOPED CONVECTIVE FLOW(FT)	NPU05210
TLHP	-	HOT	PIPE	LENGTH(FT)		NPU03800
TLHP	-	CONVERSION	OF	TLHP	TO S.I. UNITS(M)	NPU03200
TLHP	-	AMMONIA	CIRC	PIPE	LENGTH(FT)	NPU03810
TLNHP	-	CONVERSION	OF	TLNHP	TO S.I. UNITS(M)	NPU03210
TLNHP	-	AMMONIA	RE-FLUX	PIPE	LENGTH(FT)	NPU03820
TLNHP	-	CONVERSION	OF	TLNHP	TO S.I. UNITS(M)	NPU03010
TNCT	-	TOTAL	NUMBER	OF	CONDENSER TUBES	NPU01000
TNCT	-	TOTAL	NUMBER	OF	EVAPORATOR TUBES	NPU04510
TP(TDOE)	-	SUBROUTINE	TO	SIZE	COND TUBE WALL THICK(IN)	NPU04200
TP(TDOE)	-	SUBROUTINE	TO	SIZE	EVAP TUBE WALL THICK(IN)	NPU04150
TR2C	-	THERMAL	RESISTANCE	FOR	SW(HR.FT2.F/8TU)	NPU02110
TR2C	-	THERMAL	RESISTANCE	FOR	SW FOULING(HR.FT2.F/8TU)	NPU02120
TR3C	-	THERMAL	RESISTANCE	FOR	WALL THICK(HR.FT2.F/8TU)	NPU02130
TR3C	-	THERMAL	RESISTANCE	FOR	AMMONIA(HR.FT2.F/8TU)	NPU02140
TR5E	-	THERMAL	RESISTANCE	FOR	EVAP SW(HR.FT2.F/8TU)	NPU04760
TR2E	-	THERMAL	RESISTANCE	FOR	EVAP SW FOULING(HR.FT2.F/8TU)	NPU05100
TR3E	-	THERMAL	RESISTANCE	FOR	WALL THICK(HR.FT2.F/8TU)	NPU04780
TR3E	-	THERMAL	RESISTANCE	FOR	AMMONIA(HR.FT2.F/8TU)	NPU05130
TRNE1	-	EVAPORATOR	TRANSITION	REYNOLDS	NUMBER	NPU05220
TRNE2	-	EVAPORATOR	TRANSITION	REYNOLDS	NUMBER	NPU05260
TRNE3	-	EVAPORATOR	TRANSITION	REYNOLDS	NUMBER	NPU04250
TSAT(PCND)	-	SUBROUTINE	FOR	SAT	AMMONIA TEMP(DEG F)	NPU04260
TSAT(PEVAP)	-	SUBROUTINE	FOR	SAT	AMMONIA TEMP(DEG F)	NPU04640
TSDE	-	EVAPORATOR	TUBE	SHEET	DIAMETER(FT)	NPU03230
TSDE	-	CONVERSION	OF	TSDE	TO S.I. UNITS(M)	NPU03220
TSDE	-	CONVERSION	OF	TSDE	TO S.I. UNITS(M)	NPU04190
TTCC	-	CONDENSER	TUBE	WALL	THICKNESS(IN)	NPU03240
TTCC	-	CONVERSION	OF	TTCC	TO S.I. UNITS(MM)	NPU04040
TTCC	-	EVAPORATOR	TUBE	WALL	THICKNESS(IN)	NPU03250
TTCC	-	CONVERSION	OF	TTCC	TO S.I. UNITS(MM)	NPU02240
TTSC	-	THICKNESS	OF	EVAP	SHEET	NPU05350
TTSC	-	THICKNESS	OF	EVAP	SHEET	





NPU005190  
 NPU000380  
 NPU000360  
 NPU000370  
 NPU002430  
 NPU005080  
 NPU005070  
 NPU001560  
 NPU005750  
 NPU005860  
 NPU005850  
 NPU005870  
 NPU000450  
 NPU000460  
 NPU000570  
 NPU000580

C WEA - AMMONIA FLOW RATE PER UNIT AXIAL LENGTH(LBM/HR. FT)  
 C WELECT - GROSS ELECTRICAL LOADING(MW)  
 C WELECT - NET ELECT OUTPUT AS A F(EFFICIENCY)(MW)  
 C WELOSS - MEASURED LOSSES IN EFFICIENCY(MW)  
 C WNET - NET WORK IN THERMODYNAMIC CIRC PUMP WORK(BTU/LBM)  
 C WPSNH3 - THERMODYNAMIC AMMONIA CIRC PUMP WORK(BTU/LBM)  
 C X3A - ISENTROPIC AMMONIA AT STATE PT 3(DECIMAL)  
 C X3 - QUALITY OF AMMONIA AT STATE PT 3(DECIMAL)  
 C X3P - QUALITY OF AMMONIA AT STATE PT 3(PERCENT)  
 C X4P - QUALITY OF AMMONIA AT STATE PT 4(PERCENT)  
 C X5 - QUALITY AT STATE PT 5(DECIMAL)  
 C X5P - CONVERSION X5(PCT)  
 C X5S - QUALITY AT STATE PT 5S(DECIMAL)  
 C X5SP - CONVERSION OF X5S(PCT)

C	THESES PRESENTATION:MODEL ANALYSIS OF THE CLOSED-CYCLE	1
C	OCEAN THERMAL ENERGY CONVERSION(OTEC)POWER SYSTEM	2
C	BY L COR RAYMOND C. SCHAUBEL,USN	3
	SUBROUTINE ANALIZ (ICALC)	4
	COMMON /GLOBCM/ DICP,DHP,DINH3,DINH3R,PEVAP,PCOND,TDOE,TLE,T	5
	1LC,VSWC,VSWCP,VSWE,VSWHP,EPR,CPR,PRSYST,DIEMP1,DIEMP2,DIEMP5,DH5,D	6
	2X5,DTLC,ETURBP,TSDE,TSDC,OBJ	7
	IF (ICALC.GT.1.) GO TO 170	8
C	CONSTANTS	9
C	BTUC=778.2	10
	CGNC=35	11
	CONH=35	12
	CMW=746.E-06	13
	G=32.174	14
	GC=32.174	15
	GAL=7.481	16
	HPC=550.	17
	PIE=3.141592654	18
C	SALT WATER TEMPERATURE INPUTS(DEG F)	19
C	TCIC=40.	20
C	THIE=80.	21
C	INITIALIZED DESIGN VARIABLES	22
	DICP=19.6	23
	DHP=19.3	24
	DINH3=2.0	25
	DINH3R=2.0	26
	PCOND=89.	27
	PEVAP=130.	28
	TDOE=1.	29
	TDOE=1.5	30
	TLC=56.5	31
	TLE=40.	32
	VSWC=6.	33
	VSWCP=5.5	34
	VSWE=6.	35
	VSWHP=4.5	36
	EPR=1.5	37
	CPR=1.5	38
C	EVAPORATOR/CONDENSER TRIAL SELECTIONS	39
C	TYPEE=1. HORIZONTAL EVAPORATOR OWENS CORRELATION	40
C	TYPEE=2. HORIZONTAL EVAPORATOR NON-BOILING CORRELATION	41
		42
		43
		44
		45
		46
		47
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49 A  
 50 A  
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 96 A

C TYPEEE=3. HORIZONTAL EVAPORATOR BOILING CORRELATION  
 C TYPEEC=1. HORIZONTAL CONDENSER  
 C TYPEEC=2. VERTICAL CONDENSER  
 C TYPEEC=1.  
 C TYPEEE=1.  
 C ENHANCEMENT SELECTION  
 C EHFE=1. PLAIN TUBE EVAPORATOR  
 C EHFC=1. PLAIN TUBE CONDENSER  
 C EHFE=2. LINDE-PROMOTER EVAPORATOR (NOT CODED)  
 C EHFC=2. LINDE-PROMOTER CONDENSER (NOT CODED)  
 C EHFC=1.  
 C EHFE=1.  
 C TUBE FIN EFFICIENCY  
 C EFFIE=1.0  
 C EFFOE=1.0  
 C EFFOC=1.0  
 C EFFIC=1.0  
 C TUBE PROFILE SELECTION  
 C PROF=1. STAGGERED EQUI-LATERAL  
 C PROF=2. IN-LINE EQUI-SIDED  
 C PROF=1.  
 C SYSTEM PIPE GEOMETRY AND MATERIAL  
 C ECC=150.E-06  
 C ECP=500.E-06  
 C ECE=150.E-06  
 C EHP=500.E-06  
 C ELBOW=30.  
 C ENH3P=500.E-06  
 C TLCP=3000.  
 C TLHP=300.  
 C TLNH3P=150.  
 C TLNHRP=50.  
 C TUBING MATERIAL SELECTION  
 C TMAIL=1. ALUMINUM  
 C TMAIL=2. TITANIUM  
 C TMAIL=1.  
 C TKW=77.

C C PROJECTED INFLATION RATE FOR AL TUBE REINSTALLATION(PCT)  
 C C AINT=10.  
 C C  
 C C EVAP/MOISTURE SEPARATOR QUALITY REQUIREMENTS(PCT)  
 C C X3P=92.  
 C C X4P=99.5  
 C C  
 C C SALT WATER FOULING COEFFICIENT(HR.FT.F/BTU)  
 C C SWFC=.00025  
 C C  
 C C PUMP AND GEN-TURB EFFICIENCIES  
 C C  
 C C EEP=96.6  
 C C EMC=98.  
 C C EME=98.  
 C C EMNH3=98.  
 C C EPNHR=98.  
 C C EPC=85.  
 C C EPE=85.  
 C C EPNH3=75.  
 C C EPNHR=75.  
 C C ETRP=99.8  
 C C  
 C C NET OUTPUT POWER REQUIRED(MW)  
 C C ELECT=30.  
 C C  
 C C INPJT DATA SI UNIT CONVERSION  
 C C  
 C C DICPC=.3048\*DICP  
 C C DIHPC=.3048\*DIHP  
 C C DINH3C=.3048\*DINH3  
 C C DINHRC=.3048\*DINH3R  
 C C PCONDC=6.89476E-03\*PCOND  
 C C PEVAPC=6.89476E-03\*PEVAP  
 C C ICICCC=5.\*(ICIC-32.)/9.  
 C C TDODCC=25.4\*TDCC  
 C C TDDECC=25.4\*TDDE  
 C C THIEC=5.\*(THIE-32.)/9.  
 C C TKWC=1.73707\*TKW  
 C C TLCC=.3048\*TL  
 C C TLEC=.3048\*TLE  
 C C TLHPC=.3048\*TLHP  
 C C TLNHPC=.3048\*TLNH3P

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TLNRPC=.3048*TLNHRP
TLCPC=.3048*TLCPC
VSWCC=.3048*VSWC
VSWEC=.3048*VSWC
VSWPC=.3048*VSWCP
VSWHPC=.3048*VSWHP

C SUMMARY OF INPUT DATA
C
10 WRITE (6,1150)
   IF (TYPE.EQ.2.) GO TO 10
   WRITE (6,1160)
   GO TO 20
   CONTINUE
20 WRITE (6,1170)
   CONTINUE
   WRITE (6,1180) TDOE, TDOECC
   WRITE (6,1190) TLE, TLEC
   WRITE (6,1200) VSWC, VSWEC
   WRITE (6,1210) PEVAP, PEVAPC
   IF (T.MATL.EQ.2.) GO TO 30
   WRITE (6,1220) TKW, TKWC
   GO TO 40
   CONTINUE
30 WRITE (6,1230) TKW, TKWC
   CONTINUE
40 IF (PROF.EQ.2.) GO TO 50
   WRITE (6,1240)
   GO TO 60
   CONTINUE
50 WRITE (6,1250)
   CONTINUE
60 WRITE (6,1260) EPR
   IF (EHFE.EQ.2.) GO TO 70
   WRITE (6,1270)
   GO TO 80
   CONTINUE
70 WRITE (6,1280)
   CONTINUE
80 IF (TYPEC.EQ.2.) GO TO 90
   WRITE (6,1290)
   GO TO 100
   CONTINUE
90 WRITE (6,1300)
   CONTINUE
100 WRITE (6,1310) TDOCC, TDOCCC
   WRITE (6,1320) TLC, TLCC
   WRITE (6,1330) VSWC, VSWCC

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WRITE (6,1340) PCOND,PCONDC  
 IF (JMATL-EQ.2.) GO TO 110  
 WRITE (6,1350) TKW,TKWC  
 GO TO 120  
 CONTINUE  
 WRITE (6,1360) TKW,TKWC  
 CONTINUE  
 IF (PROF-EQ.2.) GO TO 130  
 WRITE (6,1370)  
 GO TO 140  
 CONTINUE  
 WRITE (6,1380)  
 CONTINUE  
 WRITE (6,1390) EPR  
 IF (EHFC-EQ.2.) GO TO 150  
 WRITE (6,1400)  
 GO TO 160  
 CONTINUE  
 WRITE (6,1410)  
 CONTINUE  
 WRITE (6,1420)  
 WRITE (6,1430) DIHP,DIHPC  
 WRITE (6,1440) TLHP,TLHPC  
 WRITE (6,1450) VSWHP,VSWHPC  
 WRITE (6,1460) THIE,THIEC  
 WRITE (6,1470) CONH  
 WRITE (6,1480)  
 WRITE (6,1490) DICP,DICPC  
 WRITE (6,1500) TLCP,TLCP  
 WRITE (6,1510) VSWCP,VSWCPC  
 WRITE (6,1520) TCIC,TCICC  
 WRITE (6,1530) CONC  
 WRITE (6,1540)  
 WRITE (6,1550) DINH3,DINH3C  
 WRITE (6,1560) TLNH3P,TLNHPC  
 WRITE (6,1570)  
 WRITE (6,1580) DINH3R,DINHRC  
 WRITE (6,1590) TLNHRP,TLNRPC  
 WRITE (6,1600)  
 WRITE (6,1610)  
 WRITE (6,1620) EPE,EME  
 WRITE (6,1630)  
 WRITE (6,1640) EPC,EMC  
 WRITE (6,1650)  
 WRITE (6,1660) EPNH3,EMNH3  
 WRITE (6,1670) EEP  
 WRITE (6,1680) ETRP  
 WRITE (6,1690)

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170      WRITE (6,1700) ELECT
      RETURN
      CONTINUE
C
C **** EXECUTION OF INPUT DATA ****
C
C TUBE WALL THICKNESS(INCH) AND TUBE COST PER FT($/FT)
C
C ALUMINUM TUBING
      IF (TMAIL.GT.1.5) GO TO 210
      IF (EHFE.GT.1.5) GO TO 180
      TTE=AP(TDOE)
      E1=AC(TDOE)
      E2=0.0
      GO TO 190
180      CONTINUE
      TTE=AE(TDOE)
      E1=AEC(TDOE)
      E2=0.0
      CONTINUE
190      CONTINUE
      IF (EHFC.GT.1.5) GO TO 200
      TTC=AP(TDOC)
      C1=AC(TDOC)
      C2=0.0
      GO TO 250
200      CONTINUE
      TTC=AE(TDOC)
      C1=ACC(TDOC)
      C2=0.0
      GO TO 250
C
C TITANIUM TUBING
C
210      CONTINUE
      IF (EHFE.EQ.2.) GO TO 220
      TTE=TP(TDOE)
      E1=TC(TDOE)
      E2=1.21
      GO TO 230
220      CONTINUE
      TTE=TE(TDOE)
      E1=TEC(TDOE)
      E2=1.21
      CONTINUE
230      CONTINUE
      IF (EHFC.EQ.2.) GO TO 240
      TTC=TP(TDOC)
      C1=TC(TDOC)

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[illegible]



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C C INITIALLY ASSUME AN EVAP TUBE LENGTH(FT)
C C ASSUME A HT COEF FOR EVAP SHELLSIDE
C C HTNH3E=1000.
C C HOT PIPE SW MASS FLOW RATE(LBM/HR)
C C
C C ROSWHP=RHOSW*(THIE)
C C FLOHP=3600.*ROSWHP*PIE*DIHP**2*VSWHP/4.
C C EVAP SW DENSITY(INITALLY ASSUME TBULK=THIE)
C C
C C TBLKE=THIE
C C CONTINUE
C C RHOSWE=RHOSW*(TBLKE)
C C TKSWE=TKSW*(TBLKE)
C C
C C TOTAL NUMBER OF EVAP TUBES
C C
C C INET=4.*FLOHP/(3600.*RHOSWE*PIE*IDIEC**2*VSWE)
C C
C C TUBE SHEET DIAMETER(FT)
C C
C C TUBE PROFILE - STAGGERED
C C
C C IF (PROF.EQ.2.) GO TO 270
C C EHT=EPR*TD0E*0.5
C C EBASE=EPR*TD0E*0.866
C C ETAREA=EHT*EBASE*2.
C C SN=2.*EHT
C C SPE=2.*EBASE-TD0E
C C SPEC=SPE/12.
C C GO TO 280
C C CONTINUE
C C
C C TUBE PROFILE - IN-LINE
C C
C C EPLONG=EPR*TD0E
C C SN=EPLONG
C C EPLAT=EPR*TD0E
C C ETAREA=EPLONG*EPLAT
C C SPE=EPR*TD0E-TD0E
C C SPEC=SPE/12.
C C CONTINUE
C C
C C EAREA=ETAREA*TNET
C C TSDE=((4.*EAREA/PIE)**0.5)/12.
C C
C C TOTAL HT AREA OF EVAP(FT2 OR M2)

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C      THTAE=INET*PIE*IDOE*TL
C      CMIN FOR EVAP(BTU/HR.F OR W/C)
C      CPSWE=CPSW(TBLKE)
C      CMINE=FLOHP*CPSWE
C      OVERALL HT COEF(BTU/HR.FT2.F OR W/M2.C)
C      TBLKER=TBLKE+.69
C      REYNOLDS NUMBER FOR EVAP TUBESIDE
C      VISSWE=VISSW(TBLKE)
C      RNSWE=3600.*RHOSWE*VSWE*IDIEC/VISSWE
C      PRANDTL NUMBER FOR EVAP TUBESIDE
C      PNSWE=CPSWE*VISSWE/TKSWE
C      HT COEF FOR EVAP TUBESIDE
C      IF (RNSWE.GT.2300.) GO TO 290
C      LAMINAR FLOW USING SIEDER-TATE CORRELATION
C      HTSWE=1.86*TKSWE*(RNSWE*PNSWE)**.3333*(IDIEC/TLE)**.3333/IDIEC
C      GO TO 300
C      CONTINUE
C      TURBULENT FLOW USING DITUS-BOELTER CORRELATION
C      (INITIALLY ASSUME TBLKE=THIE)
C      HTSWE=.023*TKSWE*RNSWE**.8*PNSWE**.4/IDIEC
C      CONTINUE
C      THERMAL RESISTANCE FOR SW(HR.FT2.F/BTU)
C      TRIE=IDOE/(EFFIE*HTSWE*IDIEC)
C      THERMAL RESISTANCE FOR SW FOULING(HR.FT2.F/BTU)
C      HTFSWE=1./SWFC
C      TR2E=IDOE/(EFFIE*HTFSWE*IDIEC)
C      THERMAL RESISTANCE FOR WALL THICKNESS(HR.FT2.F/BTU)

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C TEST FOR A SAT TBULK TEMPERATURE
C
  DBLKT=ABS(TBLKE-RTBLKE)
  SCALB=ABS(TBLKE)
  IF (SCALB.LT.0.1) SCALB=0.1
  DELB=DBLKT/SCALB
  IF (DELB.LT.0.001) GO TO 310
  TBLKE=RTBLKE
  GO TO 260
  CONTINUE
  TBLKE=RTBLKE
310
C FILM TEMP FOR PROPERTY EVALUATION
C INITIALLY ASSUME T3(IDEAL)=T3(ACTUAL)
C
  THERMAL RESISTANCES FOR SINGLE TUBE CONDUCTANCE(UA)
  (BTU/HR.F)
  AO=PIE*IDOE*ILE
  THERMAL RESISTANCE SW(HR.F/BTU)
  CTR1E=TR1E/AO
  THERMAL RESISTANCE SW FOULING(HR.F/BTU)
  CTR2E=TR2E/AO
  THERMAL RESISTANCE WALL THICKNESS(HR.F/BTU)
  CTR3E=TR3E/AO
  THERMAL RESISTANCE FOR NH3 FOULING(HR.F/BTU)
  NEGLIGIBLE
  THERMAL RESISTANCE NH3(HR.F/BTU)
  CTR5E=TR5E/AO
  HEAT TRANSFERED PER TUBE(8TU/HR)
  QET=(TBLKE-T3)/(CTR1E+CTR2E+CTR3E+CTR5E)
  SHELLSIDE WALL TEMPERATURE(F)
  ETW2=TBLKE-QET*(CTR1E+CTR2E+CTR3E)
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C C TUBESIDE WALL TEMPERATURE(F)
C C ETW1=TBLKE-QET*(CTR1E+CTR2E)
C C EVAP FILM TEMP CALCULATION(F)
C C EFT=(ETW2+T3)/2.
C C DELTA T TEMPERATURE(F)
C C DELTAE=ETW2-T3
C C AMOUNT OF HEAT ADDITION(BTU/HR OR W)
C C QE=CMINE*(THIE-THOE)
C C LOG MEAN TEMPERATURE DIFFERENCE OF EVAP(F OR C)
C C INITIALLY ASSUME T3(IDEAL)=T3(ACTUAL)
C C ELMTD=(1.-EXP(-ENTU))*CMINE*(THIE-T3)/(UE*THTAE)
C C ISENTROPIC NH3 PUMP WORK(BTU/LBM)
C C INITIALLY ASSUME P1(IDEAL)=P1(ACTUAL)
C C VF1=1./RFNH3(T1)
C C WPSNH3=VF1*(PEVAP-P1)*144./BTUC
C C THERMODYNAMIC NH3 PUMP WORK(BTU/LBM)
C C EPNH3C=EPNH3/100.
C C WPNH3=WPSNH3/EPNH3C
C C WORKING FLUID PROPERTIES
C C CPNH3E=CPNH3(EFT)
C C VSNH3E=VSNH3(EFT)
C C TKNH3E=TKNH3(EFT)
C C RONH3E=RFNH3(EFT)
C C ENTHALPY AT STATE PT 2(BTU/LBM)
C C H2=H1+WPNH3
C C ENTHALPY AT STATE PT 3A(BTU/LBM)
C C H3A=HF(PEVAP)
C C INITIALIZE NH3 MASS FLOW RATE(LBM/HR)

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C      H3=HG(PEVAP)
      FLONH3=QE/(H3-H2)
      CONTINUE
320
C      PRESS DROP EVAP SHELLSIDE(LBF/IN2)
C      ASSUME VISCOSITY(TWALL)=VISCOSITY(TBULK)
C      MAX VELOCITY THRU MIN-FLOW AREA F(TUBE PROFILE)
C      RONH3P=RFNH3(T1)
      VNH3E=4.*FLONH3/(3600.*RONH3P*PIE*DINNH3**2)
      VMAXE=VNH3E*(SN/(SN-TDOE))
C      REYNOLDS NO. FOR MAX SHELLSIDE FLOW
      REMAX=3600.*RONH3E*VMAXE*TDOE/(12.*VSNH3E)
C      EMPIRICAL FRICTION FACTOR USING CORRELATION BY JAKOB
      IF (PROF.EQ.2.) GO TO 330
      EFF=10.25+0.118/((SN-TDOE)/TDOE)**1.08)*REMAX**(-.16)
      GO TO 340
330
      CONTINUE
      EFF=10.44+(0.08*SN/TDOE)/((SN-TDOE)/TDOE)**(0.43+1.13*TDOE/SN))*RE
      MAX**(-.15)
340
      CONTINUE
      MASS VELOCITY FOR MIN FREE-FLOW AREA(LBM/FT2.SEC)
      EAF=TSOE*TLE
      EAF=EAF*(1/(SN-TDOE)/SN)
      EGF=FLONH3/(3600.*EAF)
C      CALCULATION OF PRESS DROP EVAP SHELLSIDE(LBF/IN2)
      USING THE HOMOGENEOUS TWO-PHASE MODEL
      X3A=1.0
      EDE=(EPR*TDOE-TDOE)/12.
      VLIQE=1./RFNH3(T3A)
      VAPE=1./RGNH3(PEVAP)
      VAVGE=VLIQE*(1.+X3A*(VAPE-VLIQE)/VLIQE)
      EFRICT=(EFF*EGF**2*VAVGE*TSOE)/(144.*GC)
      EMOM=(EGF*TSOE)/(144.*GC)
      EELEV=(EGF*TSOE)/(144.*GC)
      DSEVAP=EFRICT*EMOM+EELEV
C      ENTHALPY AT STATE PT 3(BTU/LBM)

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C C ASSUME QUALITY EVAPORATOR OUTLET X3P
C
X3=X3P/100.
P3=PEVAP-DSEVAP
H3F=HF(P3)
H3G=HG(P3)
H3=H3F+X3*(H3G-H3F)
C
C PRESSURE DROP ACROSS THE MOISTURE SEPARATOR(LBF/IN2)
C
RONH3S=RGNH3(P3)
ESPACE=0.1*TSDE*TLE
VNH3S=FLOH3/(3600.*RONH3S*ESPACE)
VHEAD=RONH3S*VNH3S**2/(2.*GC*144.)
DSEDEM=20.*VHEAD
C
C MOISTURE SEPARATOR DISCHARGE DRAIN ENTHALPY(BTU/LBM)
C
P4=P3-DSEDEM
T4=TSAT(P4)
PAVGDE=(P3+P4)/2.
HDE=HF(PAVGD)
C
C ENTHALPY AT STATE PT 4(BTU/LBM)
C
C ASSUME QUALITY OF MOISTURE SEPARATOR OUTLET X4P
C
X4=X4P/100.
H4F=HF(P4)
H4G=HG(P4)
H4=H4F+X4*(H4G-H4F)
C
C REVISED AMMONIA FLOW RATE(LBM/HR)
C
FLOH3=QE/((X4/X3)*H3-H2-(X4/X3-1.)*HDE)
C
C REVISED AMMONIA SHELLSIDE VELOCITY(FT/SEC)
C
VNH3ER=4.*FLOH3/(3600.*RONH3P*PIE*DINH3**2)
C
C TEST FOR SAT VNH3E(FT/SEC)
C
DVNH3E=ABS(VNH3ER-VNH3E)
SCALV=ABS(VNH3ER)
IF (SCALV.LT.0.1) SCALV=0.1
DELV=DVNH3E/SCALV
IF (DELV.LT.0.001) GO TO 350
VNH3E=VNH3ER
GO TO 320

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350 CONTINUE
   C VNH3E=VNH3ER
   C REVISED TEMP AT STATE PT 3(DEG F)
   C
   C T3R=TSAT(P3)
   C TEST FOR SAT T3(DEG F)
   C
   C DTEMP3=ABS(T3R-T3)
   C SCALT=ABS(T3R)
   C IF (SCALT.LT.0.1) SCALT=0.1
   C DELT3=DTEMP3/SCALT
   C IF (DELT3.LT.0.001) GO TO 360
   C T3=T3R
   C GO TO 260
360 CONTINUE
   C T3=T3R
   C PRANDTL NUMBER
   C
   C PNH3E=CPNH3E*VSNH3E/TKNH3E
   C EVAPORATOR TYPE IDENTIFICATION
   C TYPEE=1:HORIZONTAL OWENS CORRELATION
   C TYPEE=2:HORIZONTAL NON-BOILING CORRELATION
   C TYPEE=3:HORIZONTAL BOILING CORRELATION
   C
   C IF (TYPEE.GT.1.) GO TO 380
   C HORIZONTAL NON-BOILING USING OWENS CORRELATION
   C
   C TRANSITION REYNOLDS NUMBER
   C TRNE1=1680.*(CPNH3E*VSNH3E/TKNH3E)**(-1.5)
   C REYNOLDS NUMBER(PSEUDO)
   C
   C WE=FLONH3/TNET
   C RNH3EH=4.*WE/(ILE*VSNH3E)
   C IF (RNH3EH.GT.TRNE1) GO TO 370
   C LAMINAR FLOW USING OWENS CORRELATION
   C
   C HNH3ER=2.2*(SPEC/IDDEC)**.1*(VSNH3E**2/(3600.**2*G*RONH3E**2*TKNH3
   C IE**3))**(-.3333)*RNH3EH**(-.3333)
   C GO TO 420
   C CONTINUE
370

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189

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      HFNH3E=HF(P3)
      HGNH3E=HG(P3)
      TENS=1.6038998E-03
      CSF=.0154
      HTB=VSNH3E*(HGNH3E-HFNH3E)/(CSF**3*(TENS/RONH3E)**.5)*(CPNH3E/(1(HG
      1NH3E-HFNH3E)*PNH3E))**3*DELTAE**2
      LAMINAR FLOW CORRELATION
      HNH3ER=HTB+HTDR+HTFDR
      GO TO 420
      CONTINUE
      TURBULENT FLOW USING LORENZ AND YUNG CORRELATIONS
      CONVECTION IN DEVELOPING REGION
      HTDR=3.*CPNH3E*WEA/TLFD
      CONVECTION IN FULLY DEVELOPED REGION
      HTFDR=3.8E-03*(VSNH3E**2/(3600.**2*RONH3E**2*TKNH3E**3*G))**(-.333
      13)*RNH3EV**4*PNH3E**65
      BOILING USING LORENZ AND YUNG CORRELATION
      HGNH3E=HG(P3)
      HFNH3E=HF(P3)
      TENS=1.6038998E-03
      CSF=.0154
      HTB=VSNH3E*(HGNH3E-HFNH3E)/(CSF**3*(TENS/RONH3E)**.5)*(CPNH3E/(1(HG
      1NH3E-HFNH3E)*PNH3E))**3*DELTAE**2
      TURBULENT FLOW CALCULATIONS
      HNH3ER=HTB+HTDR+HTFDR
      CONTINUE
      TEST FOR SAT HTNH3E
      DHTNH3=ABS(HNH3ER-HTNH3E)
      SCALH=ABS(HNH3ER)
      IF (SCALH.LT.0.1) SCALH=0.1
      DELH=DHTNH3/SCALH
      IF (DELH.LT.0.001) GO TO 430
      HTNH3E=HNH3ER
      GO TO 260
      CONTINUE

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      HTNH3E=HNH3ER
C  REVISED EVAP HEAT TRANSFER AREA(FT2)
C
      THTAER=ENTU*CMINE/UE
C  REVISED EVAP TUBE LENGTH(FT)
C
      TLER=THTAER/(PIE*TD0EC*TNET)
C  TEST FOR SAT TUBE LENGTH
C
      COST CF EVAPORATOR UNIT($ )
      IF (TSDE.GT.35.) GO TO 480
C  EVAPORATOR TUBE SHEET DIAMETER(10-35)FT
C
      DRILLING TIME/TUBE SHELL THICK(MIN/IN)
C
      DTE=0.66*(TD0E-.5)
C  THICKNESS OF TUBE SHEET(IN)
C
      TTSE=0.56*TSDE**0.68
C  TUBE SHEET LABOR COST($ )
C
      CTSLE=156695.*(TNET/9630.)*(DTE/0.66)*(TTSE/4.)
C  TUBE SHEET MATERIAL COST($ )
C
      CTSME=189.486*TSDE**2.3
      IF (TMATL.EQ.1.) GO TO 440
C  TUBE MATERIAL COST($ )
C
      CTME=(E1*ILE+E2)*TNET*(TD0E/1.5)
C  TUBE INSTALLATION COST($ )
C
      CTIE=34.*TNET*TD0E**0.7
      GO TO 450
      CONTINUE
C  TUBF MATERIAL COST($ )
C
      CTME=(E1*ILE+E2)*TNET*(TD0E/1.5)*(1.+(1.+AINT/100.))**10+(1.+AINT/1
      100.))**20)
C
      440
C
      C
      C

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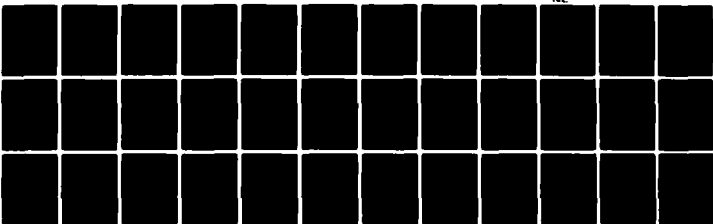
NAVAL POSTGRADUATE SCHOOL MONTEREY CA  
OPTIMIZATION OF A LOW DELTA T RANKINE POWER SYSTEM.(U)  
DEC 80 R C SCHAUBEL

F/G 20/13

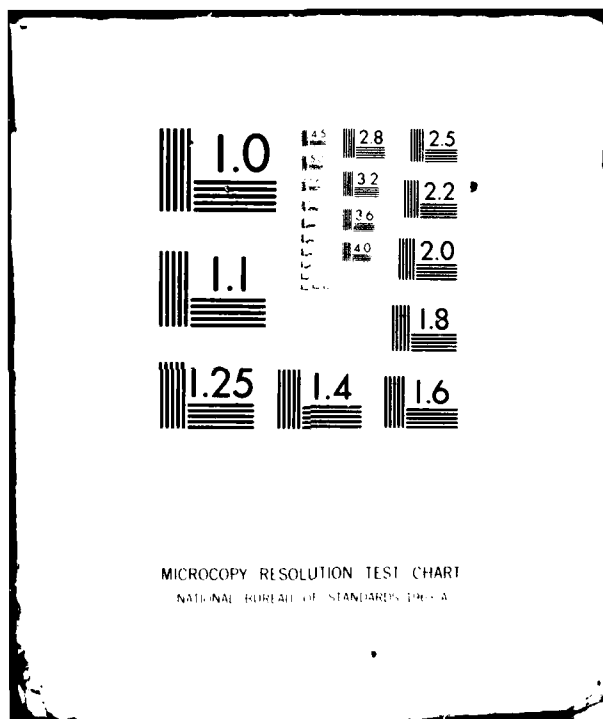
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C	TTSE=0.56*TSDE**0.68	A 961
C	TUBE SHEET MATERIAL AND LABOR COSTS(\$)	A 962
C	IF (TMAIL.EQ.1.) GO TO 490	A 963
	CTSL=55.189*TNET**0.791*TSDE**0.68*DTE	A 964
	CTSM=29.566*TSDE**2.014*TTSE	A 965
	GO TO 500	A 966
490	CONTINUE	A 967
	CTSL=73.81*TNET**0.791*TSDE**0.68*DTE	A 968
	CTSM=354.3*TSDE**1.61*TTSE	A 969
500	CONTINUE	A 970
	IF (TMAIL.EQ.1.) GO TO 510	A 971
C	TUBE MATERIAL COST(\$)	A 972
C	CTME=(E1*TLE+E2)*TNET*(TDOE/1.5)	A 973
C	TUBE INSTALLATION COST(\$)	A 974
C	CTIE=36.542*TNET*TDOE**0.7	A 975
510	GO TO 520	A 976
C	CONTINUE	A 977
	TUBE MATERIAL COST(\$)	A 978
C	CTME=(E1*TLE+C2)*TNET*(TDOE/1.5)*(1.+(1.+AINT/100.))**10+(1.+AINT/100.)*20)	A 979
C	TUBE INSTALLATION COST(\$)	A 980
	CTIE=36.542*TNET*TDOE**0.7	A 981
520	GO TO 520	A 982
C	CONTINUE	A 983
	TUBE MATERIAL COST(\$)	A 984
C	CTME=(E1*TLE+C2)*TNET*(TDOE/1.5)*(1.+(1.+AINT/100.))**10+(1.+AINT/100.)*20)	A 985
C	TUBE INSTALLATION COST(\$)	A 986
	CTIE=36.542*TNET*TDOE**0.7*(1.+(1.+AINT/100.))**10+(1.+AINT/100.)*20)	A 987
520	CONTINUE	A 988
	HEAT EXCH SHELL COST(\$)	A 989
C	CHSE=12.544*(TLE+6.)*TSDE**2.06	A 990
C	NH3 DIST PLATE AND BAFFLES COST(\$)	A 991
	CDPBE=158.099*TSDE**1.82+72.419*TNET**0.873*DTE	A 992
C	BUSTLE, FLANGE, CHANNELS AND FLOW PLATE COSTS(\$)	A 993
C	CBFCFE=472.977*TSDE**2.12	A 994
C	HEAT EXCH HEAD COST(\$)	A 995
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C      CHE=1725.31*TSDE**1.45
C      WATER INLET,NOZZLES AND SUPPORTS COST($)
C      CWINSE=7445.297*TSDE**1.1
C      TUBE WELDING COSTS($)
C      CTWE=0.0
C      IF (THATL.EQ.1.) GO TO 540
C      IF (TNET.GT.36000) GO TO 530
C      CTWE=14.73*TNET**1.03*(TDOE/1.5)**0.7
C      GO TO 540
C      CONTINUE
530    CTWE=0.8797*TNET**1.3*(TDOE/1.5)**0.7
540    CONTINUE
550    CEVAP=(CTSLE+CTSME+CTME+CTIE+CHSE+CDPBE+CBFCFE+CHE+CWINSE+CTWE)
C      *****
C      *
C      * PARASITIC LOSS SECTION
C      *
C      *
C      *****
C      EVAPORATOR SALT WATER PUMP OR HOT PIPE PUMP
C      DELTA P EVAP SW PUMP
C      ROSWHP=RHOSW*(THIE)
C      THIER=THIE+459.69
C      VISWHP=VISSW*(THIE)
C      DELTA P SW HOT PIPE USING DARCEY-WEISBACH CORRELATION
C      (LBF/IN2)
C      REYNOLDS NUMBER FOR HOT PIPE FLOW
C      RNSWHP=3600.*ROSWHP*VSWHP*DIHP/VISWHP
C      FRICTION FACTOR FOR LAMINAR FLOW
C      IF (RNSWHP.GT.2300.) GO TO 560
C      FFHP=64./RNSWHP
C      GO TO 570

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A	A	A	A	A	A	A	A	A	A	A	A	A	A	A	A	A	A	A	A
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0	0	6	7	8	9	0	1	2	3	4	5	6	7	8	9	0	1	2	3
5	0	6	7	8	9	0	1	2	3	4	5	6	7	8	9	0	1	2	3

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C DARCEY-WEISBACH CORRELATION(LBF/IN2)
C
C REYNOLDS NUMBER FOR COLD PIPE FLOW
C
C RNSWCP=3600.*ROSWCP*VSWCP*DICP/VISWCP
C
C FRICTION FACTOR LAMINAR FLOW
C
C IF (RNSWCP.GT.2300.) GO TO 600
C FFCP=64./RNSWCP
C GO TO 610
C CONTINUE
C
C FRICTION FACTOR TURBULENT FLOW USING STREETER CORR
C
C FFCP=1.325/(ALOG(ECP/(3.7*DICP))+5.74/RNSWCP**2)**2
C CONTINUE
C
C MINOR ENTRY/EXIT LOSSES(ASSUME KI=.05 WELL ROUNDED
C PIPE ENTRANCE AND ENTERS PLENUM PRIOR TO CONDENSER
C WHERE AREA IS ABRUPTLY CHANGED TO 2 TIMES PIPE
C DIAMETER)(LBF/IN2)
C
C CPKI=.05
C CPKE=(1.-(DICP/(2.*DICP))**2)**2
C DDUCTC=(CPKI+CPKE)*ROSWCP*VSWCP**2/(2.*GC*144.)
C
C PIPE FRICTION LOSSES(LBF/IN2)
C
C RKCP=FFCP*(TLCF/DICP+EL80W)
C DPIPEC=RKCP*ROSWCP*VSWCP**2/(2.*GC*144.)
C
C DELTA P PIPE LOSS CALCULATION(LBF/IN2)
C
C DPIPCC=DDUCTC+DPIPEC
C
C DELTA P DUE TO SM DENSITY(LBF/IN2)
C
C RDEPTH=RHOSWD(TLCF)
C RAVG=(RDEPTH+64.184)/2.
C DHEAD=((RDEPTH-64.184)/RDEPTH)*TLCF/2.
C DDENC=G*RAVG*DHEAD/(144.*GC)
C
C DELTA P CONDENSER(ASSUME NO OUTLET PIPING) USING
C DARCEY-WEISBACH CORRELATION(LBF/IN2)
C
C INITIALLY ASSUME TBLK=TSW(IN)
C TBLKC=TCIC

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620	CONTINUE	AI201
	TBLKCR=TBLKC+459.69	AI202
C	RHOSWC=RHOSW(TBLKC)	AI203
C	MINOR ENTRY/EXIT LOSSES(ASSUME KI=.05 WELL ROUNDED	AI204
C	TUBE ENTRANCE AND KE=1. EXPANSION TO AN INFINITE	AI205
C	RESEVOIR(LBF/IN2)	AI206
		AI207
	RKI=.05	AI208
	RKE=1	AI209
	DPHINC=(RKI+RKE)*RHOSWC*VSWC**2/(2.*GC*144.)	AI210
C	DELTA P CONDENSER CORE(LBF/IN2)	AI211
C		AI212
C		AI213
C		AI214
C		AI215
C	REYNOLDS NUMBER FOR COND TUBESIDE	AI216
		AI217
	VISSWC=VISSW(TBLKC)	AI218
C	RNSWC=3600.*RHOSWC*VSWC*TDICC/VISSWC	AI219
		AI220
	FRICTION FACTOR LAMINAR FLOW	AI221
		AI222
	IF (RNSWC.GT.2300.) GO TO 630	AI223
	FFC=64./RNSWC	AI224
630	GO TO 640	AI225
	CONTINUE	AI226
640	FFC=1.325/(ALOG(ECC/(3.7*TDICC))+5.74/RNSWC**9))**2	AI227
C	CONTINUE	AI228
C		AI229
C	DELTA P COND CORE CALCULATION(LBF/IN2)	AI230
		AI231
	RKHC=FFC*TLC/TDICC	AI232
C	DCOREC=RKHC*RHOSWC*VSWC**2/(2.*GC*144.)	AI233
C		AI234
	DELTA P CONDENSER DUE TO CORE ELEVATION(LBF/IN2)	AI235
		AI236
	IF (TYPEC.GT.1.) GO TO 650	AI237
	DPELEC=0.	AI238
	GO TO 660	AI239
	CONTINUE	AI240
650		AI241
C	VERTICAL TUBED CONDENSER SW INLET ENTERS AT THE TOP	AI242
C	OF THE HEAT EXCHANGER	AI243
C		AI244
	DPELEC=0.	AI245
660	CONTINUE	AI246
C		AI247
	DELTA P CONDENSER CALCULATION(LBF/IN2)	AI248

C DCOND=DPMINC+DCOREC+DPELEC  
 C  
 C DELTA P COND SW PUMP CALCULATION(LBF/IN2 OR FT)  
 C  
 C DPMPC=DDUCTC+DDENC+DPMINC+DCOREC+DPIPEC  
 C DPMPC=144.\*DPMPC\*GC/(RDS\*CP\*G)  
 C  
 C POWER COND SW PUMP(HP)  
 C  
 C EPCC=EPC/100.  
 C PWRCP=FLOCP\*DPMPC\*G/(EPCC\*HPC\*3600.\*GC)  
 C  
 C POWER COND SW PUMP MOTOR(MW)  
 C  
 C EMCC=EMC/100.  
 C PWRCPH=PWRCP\*CMW/EMCC  
 C  
 C DISCHARGE RATE OF COND SW PUMP(FT3/SEC OR GAL/MIN)  
 C  
 C QCPMP=PIE\*DICP\*\*2\*VSNCP/4.  
 C QCPMP=QCPMP\*60.\*GAL  
 C  
 C COST OF COND SW PUMP(\$)  
 C  
 C CCSWP=((QCPMP/1000.)\*0.75+50.)\*1.21E+03  
 C  
 C AMMONIA CIRCULATION PUMP  
 C  
 C DELTA P NH3 CIRC PUMP  
 C VSNH3P=VSNH3(T1)  
 C RHNH3P=RFNH3(T1)  
 C  
 C DELTA P NH3 PUMP(ASSUME V1=V2)USING DARCEY-WEISBACH  
 C CORRELATION(LBF/IN2)  
 C  
 C NH3 PIPE FLOW VELOCITY(FT/SEC)  
 C  
 C VNH3P=4.\*FLONH3/(3600.\*RHNH3P\*PIE\*DINH3\*\*2)  
 C  
 C REYNOLDS NUMBER FOR NH3 PIPE FLOW  
 C  
 C RNH3P=3600.\*RHNH3P\*VNH3P\*DINH3/VSNH3P  
 C  
 C FRICTION FACTOR LAMINAR FLOW  
 C  
 C IF (RHNH3P.GT.2300.) GO TO 670  
 C FFNH3P=64./RHNH3P

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670 GO TO 680
C CONTINUE
C
C FRICTION FACTOR TURBULENT FLOW USING STREETER CORR
C
C FFNH3P=1.325/(ALOG(ENH3P/(3.7*DINH3)+5.74/RNH3P**9)))**2
680 CONTINUE
C
C PIPE FRICTION LOSSES(LBF/IN2)
C
C RKNH3P=FFNH3P*(TLNH3P/DINH3+4.*ELBOW)
C DPNH3=RKNH3P*RONH3P*VNH3P**2/(2.*GC*144.)
C
C DELTA P CORE EVAPORATOR
C
C DPCORE=DSEVAP+DSDEN
C
C DELTA P DUE TO PIPING ELEVATION(LBF/IN2)
C
C EZ1=0.
C EZ2=TSDE+25
C DPELEV=RONH3P*G*(EZ2-EZ1)/(GC*144.)
C
C
C DELTA P PIPE LOSS CALCULATION(LBF/IN2)
C DPIP=DPNH3+DPELEV
C
C DELTA P THERMODYNAMICALLY(LBF/IN2)
C
C DPOTHER=PEVAP-PCOND
C
C DELTA P NH3 CIRC PUMP(LBF/IN2 OR FT)
C
C DPMN=DPNH3+DPCORE+DPELEV+DPOTHER
C DMPNC=144.*DPMN*GC/(RONH3P*G)
C
C POWER NH3 CIRC PUMP(HP)
C
C EPNH3C=EPNH3/100.
C PWRNP=FLONH3*DPMNC*G/(EPNH3C*HPC*3600.*GC)
C
C POWER NH3 CIRC PUMP MOTOR(MW)
C
C EMNH3C=EMNH3/100.
C PWRNPM=PWRNP*CMW/EMNH3C
C
C DISCHARGE RATE NH3 CIRC PUMP(FT3/SEC OR GAL/MIN)

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QNPMP=PIE*0.1NH3**2*VNH3P/4.
QNPMP=QNPMP*60.*GAL
C COST OF NH3 CIRC PUMP($ )
C
VF1=1./RFNH3(T1)
CNH3P=(FLONH3*VF1/80100.)**0.64*1.21E+05
C EVAPORATOR RE-FLUX PUMP
C ASSUME RE-FLUX MASS FLOW RATE=0.3 X FLONH3
C
DELTA P NH3 RE-FLUX PIPING(ASSUME V1=V2)
C USING DARCEY-WEISBACH CORRELATION(LBF/IN2)
PAVGE=(PEVAP+P3)/2.
TAVGE=T SAT(PAVGE)
RONH3R=RFNH3(TAVGE)
VSNHRP=VSNH3(TAVGE)
C NH3 FLOW VELOCITY(FT/SEC)
C
FLONHR=0.3*FLONH3
VNHRP=4.*FLONHR/(3600.*RONH3R*PIE*DINH3R**2)
C REYNOLDS NO FOR NH3 RE-FLUX PIPE FLOW
C
RNHRP=3600.*RONH3R*VNHRP*DINH3R/VSNHRP
C FRICTION FACTOR LAMINAR FLOW
C
IF (RNHRP.GT.2300.) GO TO 690
FNHRP=64./RNHRP
GO TO 700
CONTINUE
C
C FRICTION FACTOR TURBULENT FLOW USING STREETEER CORR
C
FNHRP=L.325/(ALOG(ENH3P/(3.7*DINH3R)+5.74/RNHRP**0.9))**2
CONTINUE
C
C PIPE FRICTION LOSSES(LBF/IN2)
C
RKNHRP=FNHRP*(TLNHRP/DINH3R+4.*ELBOW)
DPNHRP=RKNHRP*RONH3R*VNHRP**2/(2.*GC*144.)
C
C DELTA P DUE TO PIPING ELEVATION(LBF/IN2)
C
EZR=TSDE+10
DPELER=RONH3R*G*(EZR-EZ1)/(GC*144.)
  
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C C DELTA P PIPE LOSSES(LBF/IN2)  
 C C DPIP NR=DPNHRP+DPELER  
 C C DELTA P NH3 RE-FLUX THERMODYNAMICALLY(LBF/IN2)  
 C C USING A SINGLE PHASE PRESSURE MODEL  
 C C RONHEV=RGNH3(PEVAP)  
 C C DPTH RF=2.\*EFF\*EGF\*\*2\*TNET\*\*0.5/(144.\*RONHEV\*GC)  
 C C DELTA P NH3 RE-FLUX PUMP(LBF/IN OR FT)  
 C C DPMNR=DPNHRP+DPELER+DPTH RF  
 C C DPPNR=144.\*DPMNR\*GC/(RONH3R\*G)  
 C C POWER NH3 RE-FLUX PUMP(HP)  
 C C EPNHRC=EPNHR/100.  
 C C PWRRP=FLONHR\*DPNRC\*G/(EPNHRC\*HPC\*3600.\*GC)  
 C C POWER NH3 RE-FLUX PUMP MOTOR(MW)  
 C C EMNHRC=EMNHR/100.  
 C C PWRRM=PWRRP\*CMW/EMNHRC  
 C C DISCHARGE RATE NH3 RE-FLUX PUMP(FT3/SEC OR GAL/MIN)  
 C C QRPMP=PIE\*DINH3R\*\*2\*VNH3RP/4.  
 C C QRPMP C=QRPMP\*60.\*GAL  
 C C COST OF NH3 RE-FLUX PUMP(\$)  
 C C VFR=1./RFNH3(TAVGE)  
 C C CNH3RP=(FLONHR\*VFR/80100.)\*0.64\*1.21E+05  
 C C PARASITIC PUMP LOSSES(MW)  
 C C PARAL=PWREPM+PWRC PM+PWRNPM+PWRRM  
 C C \*\*\*\*\*  
 C C \*  
 C C \* TURBINE AND ELECTRICAL POWER SECTION \*  
 C C \*  
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 C C \*\*\*\*\*

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C	GROSS ELECTRICAL LOAD(MW)	A1441
C	NET ELECTRICAL OUTPUT DESIRED(ELECT- PROVIDED IN INITIAL PARAMETERS)(MW)	A1442
C	ELECTRICAL LOADING AS EFFECTED BY EFFICIENCY(MW)	A1443
C	EEPC=EEP/100.	A1444
C	ETRPC=ETRP/100.	A1445
C	WELECT=ELECT/(EEPC*ETRPC)	A1446
C	WELOSS=WELECT-ELECT	A1447
C	GROSS ELECT LOADING INCL PARASITIC LOSSES(MW)	A1448
C	WELECG=WELECT+PARAL	A1449
C	POWER GENERATOR-TURBINE(HP)	A1450
C	PWRTR=1341.*WELECG	A1451
C	TURBINE EFFICIENCY REQUIREMENT(PCT)	A1452
C	ENTHALPY AT STATE PT 5(BTU/LBM)	A1453
C	H5=H4-(3412.2E+03*WELECG/FLONH3)	A1454
C	CONSTRAINT FOR SAT STATE PT 5	A1455
C	H5G=HG(PCOND)	A1456
C	DH5=H5G-H5	A1457
C	QUALITY OF NH3 AT STATE PT5(PCT)	A1458
C	H5F=HF(PCOND)	A1459
C	X5=(H5-H5F)/(H5G-H5F)	A1460
C	X5P=X5*100.	A1461
C	QUALITY OF NH3 EXHAUST AT STATE PT 5S(PCT)	A1462
C	S4F=SF(T4)	A1463
C	S4G=SG(T4)	A1464
C	S4=X4F+X4*(S4G-S4F)	A1465
C	S5F=SF(T5)	A1466
C	S5G=SG(T5)	A1467
C	X5S=(S4-S5F)/(S5G-S5F)	A1468
C	X5SP=X5S*100.	A1469
C		A1470
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C C CONSTRAINT FOR A SAT QUALITY AT STATE PT 5S
C C DX5=X5-X5S
C C ENTHALPY AT STATE PT 5S
C C H5S=H5F+X5S*(H5G-H5F)
C C TURBINE EFFICIENCY CALCULATION(PCT)
C C ETURB=(H4-H5)/(H4-H5S)
C C ETURBP=ETURB*100.
C C COST OF NH3 TURBINE- GENERATOR($)
C C FPF=1.447
C C DF=2.0
C C CTURB=(0.375+(WELECG*1000.)/(136000*DF))*FPF*2.42E+06
C C CGEN=(WELECG*.023+.3)*1.21E+06
C C CELECT=CGEN+CTURB

C C *****
C C *
C C * CONDENSER SECTION *
C C *
C C *
C C *****

C C AMOUNT OF HEAT REJECTION(BTU/HR OR MW)
C C QC=FLONH3*(H5-H1)
C C CMIN FOR CONDENSER(BTU/HR.F)
C C CPSWC=CPSW(TBLKC)
C C CMINC=FLOCP*CPSWC
C C COND SW OUTLET TEMP(F OR C)
C C TCOC=TCIC+QC/CMINC
C C TOTAL NUMBER OF CONDENSER TUBES
C C RHOSWC=RHOSW(TBLKC)
C C TNCT=4.*FLOCP/(3600.*RHOSWC*PIE*TDICC**2*VSWC)
C C
  
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C TUBE SHEET DIAMETER(FT)
C
C TUBE PROFILE - STAGGERED
C
  IF (PROF-EQ.2.) GO TO 710
  CHT=CPR*TDCC*0.5
  CBASE=CPR*TDCC*0.866
  CTAREA=CHT*CBASE*2.
  SNC=2.*CHT
  GO TO 720
  CONTINUE
710
C
C TUBE PROFILE - IN-LINE
C
  CPLONG=CPR*TDCC
  CPLAT=CPR*TDCC
  CTAREA=CPLONG*CPLAT
  CONTINUE
720
C
  CAREA=CTAREA*TNCT
  TSOC=((4.*CAREA/PIE)*0.5)/12.
C LOG MEAN TEMP DIFFERENCE OF CONDENSER(F OR C)
C
  CLMTD=((T1-TCOC)-(T1-TCIC))/ALOG((T1-TCOC)/(T1-TCIC))
C COND SW AVG BULK TEMP(F)
C
  RTBLKC=(TCOC+TCIC)/2.
C TEST FOR SAT TBLKC
C
  DBLKTC=ABS(TBLKC-RTBLKC)
  SCALBC=ABS(TBLKC)
  IF (SCALBC.LT.0.1) SCALBC=0.1
  DELBC=DBLKTC/SCALBC
  IF (DELBC.LT.0.001) GO TO 730
  TBLKC=RTBLKC
  GO TO 620
730
C
  CONTINUE
  TBLKC=RTBLKC
  TBLKCR=TBLKC*459.69
C CONDENSER CONDUCTANCE(BTU/HR.F)
C
  UAC=QC/CLMTD
C
  C NUMBER OF TRANSFER UNITS FOR COND(NTU)
C

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C CNTU=UAC/CMINC
C COND EFFECTIVENESS(EPSILON)
C EPSC=1.-EXP(-CNTU)
C INITIALLY ASSUME A TLC
C OVERALL HEAT TRANSFER COEFFICIENT(BTU/HR.FT2.F OR W/M2)
C INITIALLY ASSUME HTNH3
C HTNH3C=1000.
C CONTINUE
C 740
C REYNOLDS NUMBER FOR COND TUBESIDE
C (PROPERTIES EVAL AT TBULK)
C VISSWC=VISSW(TBLKC)
C RHOSHR=RHOSH(TBLKC)
C RNSWC=3600.*RHOSHR*VSWC*TDICC/VISSWC
C PRANDTL NUMBER FOR COND TUBESIDE
C CPSWC=CPSW(TBLKC)
C TKSWC=TKSW(TBLKC)
C PNSWC=CPSWC*VISSWC/TKSWC
C HEAT TRANSFER COEF FOR COND SW TUBESIDE
C (BTU/HR.FT2.F OF W/M2)
C IF (RNSWC.GT.2300.) GO TO 750
C LAMINAR FLOW USING SEIDER-TATE CORRELATION(ASSUME
C VISCOSITY(TBULK)=VISCOSITY(TWALL))
C HTSWC=1.86*TKSWC*((RNSWC*PNSWC)**.3333)*(TDICC/TLC)**.3333/TDICC
C GO TO 760
C CONTINUE
C 750
C TURBULENT FLOW USING DITTUS-BOELTER CORRELATION
C HTSWC=.023*TKSWC*RNSWC**.8*PNSWC**.3/TDICC
C CONTINUE
C 760
C FILM TEMP FOR PROPERTY EVALUATION
C THERMAL RESISTANCES FOR SINGLE TUBE CONDUCTANCE
C UA - OUTSIDE(BTU/HR.F)

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C C THERMAL RESISTANCE SW(HR.F/BTU)  
 C C CTR1C=1/(EFFIC\*HTSWC\*PIE\*TDICC\*TLC)  
 C C THERMAL RESISTANCE FOR SW FOULING(HR.F/BTU)  
 C C HTFSWC=1./SWFC  
 C C CTR2C=1./(EFFIC\*HTFSWC\*PIE\*TDICC\*TLC)  
 C C THERMAL RESISTANCE FOR WALL THICKNESS(HR.F/BTU)  
 C C CTR3C=ALOG(TDOCC/TDICC)/(2.\*PIE\*TKW\*TLC)  
 C C THERMAL RESISTANCE FOR NH3 FOULING(HR.F/BTU)  
 C C (CONSIDERED NEGLIGIBLE)  
 C C THERMAL RESISTANCE FOR NH3(HR.F/BTU)  
 C C CTR5C=1./(EFFOC\*HTNH3C\*PIE\*TDGCC\*TLC)  
 C C HEAT TRANSFERED PER TUBE(BTU/HR)  
 C C QCT=(T1-TBLKC)/(CTR1C+CTR2C+CTR3C+CTR5C)  
 C C TUBE SIDE WALL TEMP(F)  
 C C CTW1=TBLKC+QCT\*(CTR1C+CTR2C)  
 C C SHELLSIDE WALL TEMP(F)  
 C C CTW2=TBLKC+QCT\*(CTR1C+CTR2C+CTR3C)  
 C C COND FILM TEMP CALCULATION(F)  
 C C CFT=(CTW2+T1)/2.  
 C C CONDENSER DELTA T TEMP(F)  
 C C DELTAC=T1-CTW2  
 C C CONO SHELLSIDE HEAT TRANSFER COEF(BTU/HR.FT2.F OR W/M2)  
 C VSNH3C=VSNH3(CFT)  
 C RONH3C=RFNH3(CFT)  
 C TKNH3C=TKNH3(CFT)



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1)*NNH3CV**4
C CONTINUE
800
C
C TEST FOR SAT HTNH3C
C
    DHNH3C=ABS(HNH3CR-HTNH3C)
    SCALC=ABS(HNH3CR)
    IF (SCALC.LT.0.1) SCALC=0.1
    DELC=DHNH3C/SCALC
    IF (DELC.LT.0.001) GO TO 810
    HTNH3C=HNH3CR
    GO TO 740
810
    CONTINUE
    HTNH3C=HNH3CR
C
C PRESSURE DROP ACROSS CONDENSER SHELLSIDE(LBF/IN2)USING
C TWO-PHASE MODEL(HOMOGENEOUS)
C
C MAX VELOCITY THRU MINIMUM- FLOW AREA F(TUBE PROFILE)
    RONH3T=RGNH3(PCOND)
    VNH3C=4.*FLONH3/(3600.*RONH3T*PIE*TSDC**2)
    VMAXC=VNH3C*(SNC/(SNC-TDOC))
C
C REYNOLDS NUMBER FOR MAXIMUM SHELLSIDE FLOW
C
    REMAXC=3600.*RONH3C*VMAXC*TDOC/(12.*VSNH3C)
C
C EMPIRICAL FRICTION FACTOR USING CORRELATION BY JAKOB
C
    IF (PROF.EQ.2.) GO TO 820
    CFF=(0.25+0.118)/((SNC-TDOC)/TDOC)**1.08*REMAXC**(-.16)
    GO TO 830
820
    CONTINUE
    CFF=(0.44+10.08*SNC/TDOC)/((SNC-TDOC)/TDOC)**(0.43+1.13*TDOC/SNC)
830
    1*REMAXC**(-.15)
    CONTINUE
C
C MASS VELOCITY FOR MINIMUM FREE-FLOW AREA(LBM/FT2.SEC)
C
    IF (TYPEC.GT.1.) GO TO 840
    CAF=TSDC*TLF
    CL=TSDC
    GO TO 850
840
    CONTINUE
    BAND=TLF
    CAF=PIE*TSDC*BAND
    CL=TLF
850
    CONTINUE

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      CAFF=CAF*((SNC-TDOCC)/SNC)
      CGF=FLONH3/(3600.*CAFF)

C  CALCULATION OF CONDENSER SHELLSIDE PRESSURE DROP(LBF/IN2)
C  USING THE HOMOGENEOUS TWO-PHASE MODEL
C
      EDC=(CPR*TDOCC-TDOCC)/12.
      VLIQC=1./RGNH3(T5)
      VAPC=1./RGNH3(PCOND)
      VAVGC=VLIQC*(1.+X5*(VAPC-VLIQC)/VLIQC)
      CFRIC=(CGF*CGF**2*VAVGC*CL)/(144.*EDC*2.*GC)
      CMOM=(CGF*CGF**2*VAVGC)/(144.*GC)
      CELEV=(CGF*CL)/(144.*GC)
      DSCOND=CFRIC+CMOM+CELEV

C  PROPERTIES AT STATE PT 1
C
      P1=PCOND-DSCOND
      H1=HF(P1)
      T1R=TSAT(P1)

C  TEST FOR SAT T1(DEG F)
C
      OTEMP4=ABS(T1R-T1)
      SCALT1=ABS(T1R)
      YF=(SCALT1-LT-0.1) SCALT1=0.1
      DELT4=DTEMP4/SCALT1
      IF (DELT4-LT-0.001) GO TO 860
      T1=T1R
      GO TO 260
      CONTINUE
      T1=T1R
860

C  THERMAL RESISTANCES FOR OVERALL HEAT TRANSFER COEF - U
C  OUTSIDE(HR.FT2.F/BTU)
C
      OUTSIDE TUBE SURFACE AREA(FT2)
      CAD=PIE*TDOCC*TLC
      THERMAL RESISTANCE FOR SW(HR.FT2.F/BTU)
      TR1C=CAO*CTR1C
      THERMAL RESISTANCE FOR SW FOULING(HR.FT2.F/BTU)
      TR2C=CAO*CTR2C
C

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C C THERMAL RESISTANCE FOR WALL THICKNESS(HR.FT2.F/8TU)  
 C TR3C=CAO\*CTR3C  
 C THERMAL RESISTANCE FOR NH3 FOULING(HR.FT2.F/8TU)  
 C (CONSIDERED NEGLIGIBLE)  
 C THERMAL RESISTANCE FOR NH3(HR.FT2.F/8TU)  
 C TR5C=CAO\*CTR5C  
 C OVERALL HEAT TRANSFER COEF U - OUTSIDE(8TU/HR.FT2.F  
 C OR W/M2)  
 C UC=1./((TR1C+TR2C+TR3C+TR5C)  
 C PSEUDO HT COEF FOR SW(8TU/HR.FT1.F OR W/M2.C)  
 C HSWC=1./TR1C  
 C PSEUDO HT COEF FOR SW FOULING(8TU/HR.FT2.F OR W/M2.C)  
 C HFSWC=1./TR2C  
 C PSEUDO HT COEF FOR WALL THICK(8TU/HR.FT2.F OF W/M2.C)  
 C HWC=1./TR3C  
 C PSEUDO HT COEF FOR NH3(8TU/HR.FT2.F OR W/M2.C)  
 C HNH3C=1./TR5C  
 C TOTAL CONDENSER HEAT TRANSFER AREA(FT2 OR M2)  
 C THTAC=CNLU\*CMINC/UC  
 C REVISED CONDENSER TUBE LENGTH(FT)  
 C TLCCR=THHTAC/(PIE\*TDCCC\*TNCT)  
 C CONSTRAINT FOR A SAT TUBE LENGTH  
 C DTLC=TLC-TLCR  
 C COST OF CONDENSER UNIT(\$)  
 C IF (TSDC.GT.35.) GO TO 910  
 C

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C	CBFCFC=1185.286*TSDC**2	AI921
C	HEAT EXCH HEAD COST(\$)	AI922
C	CHC=53240.*(TSDC/18.1)**3	AI923
C	WATER INLETS,NOZZLES AND SUPPORTS COST(\$)	AI924
C	CWNSC=10106.475*TSDC	AI925
C	TUBE WELDING COST(\$)	AI926
C	CTWC=0.0	AI927
C	IF (TMATL.EQ.1.) GO TO 900	AI928
C	IF (TNCT.GT.36000.) GO TO 890	AI929
C	CTWC=14.73*TNCT**1.03*(TDOC/1.5)**0.7	AI930
C	GO TO 900	AI931
C	CONTINUE	AI932
890	CTWC=0.8797*TNCT**1.3*(TDOC/1.5)**0.7	AI933
900	CONTINUE	AI934
	CCOND=((CTSLC+CTSMC+CTMC+CTIC+CHSC+CDPBC+CBFCFC+CHC+CWNSC+CTWC)	AI935
	GO TO 980	AI936
	CONTINUE	AI937
910		AI938
C	CONDENSER TUBE SHEET DIAMETER(35-50)FT	AI939
C	DRILLING TIME/TUBE SHELL THICKNESS(MIN/IN)	AI940
C	DTC=0.66*(TDOC-0.5)	AI941
C	THICKNESS OF TUBE SHEET(IN)	AI942
C	TTSC=0.56*TSDC**0.68	AI943
C	TUBE SHEET MATERIAL AND LABOR COST(\$)	AI944
C	IF (TMATL.EQ.1.) GO TO 920	AI945
C	CTSLC=55.189*TNCT**0.791*TSDC**0.68*DTC	AI946
C	CTSMC=29.566*TSDC**2.014*TTSC	AI947
C	GO TO 930	AI948
C	CONTINUE	AI949
920	CTSLC=73.81*TNCT**0.791*TSDC**0.68*DTC	AI950
930	CTSMC=354.3*TSDC**1.61*TTSC	AI951
C	CONTINUE	AI952
C	IF (TMATL.EQ.1.) GO TO 940	AI953
C	TUBE MATERIAL COST(\$)	AI954





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IDUCCC=25.4*IDOC
TDOECC=25.4*TDOE
THIEC=5.*(THIE-32.)/9.
THOEC=5.*(THOE-32.)/9.
THTACC=9.2903E-02*THTAC
THAE=9.2903E-02*THTAE
TKWC=1.73707*TKW
ILCC=.3048*ILC
ILCPC=.3048*TLCP
TLEC=.3048*TLE
TLHPC=.3048*TLHP
TLNHPC=.3048*TLNH3P
TSDCC=0.3048*TSDC
TSECC=0.3048*TSDE
TTCC=25.4*TTC
TTTEC=25.4*TTTE
UEC=5.6782*UE
VSWCC=.3048*VSWC
VSWEC=.3048*VSWE
VSWCPC=.3048*VSWCP
VSWHPC=.3048*VSWHP

```

C SUMMARY OF INPUT/OUTPUT DATA

```

C
WRITE (6,1720)
IF (TYPE.EQ.2.) GO TO 990
WRITE (6,1730)
GO TO 1000
990 CONTINUE
1000 CONTINUE
WRITE (6,1740)
WRITE (6,1750)
WRITE (6,1760)
WRITE (6,1770)
WRITE (6,1780)
WRITE (6,1790)
WRITE (6,1800)
WRITE (6,1810)
WRITE (6,1820)
WRITE (6,1830)
WRITE (6,1840)
WRITE (6,1850)
WRITE (6,1860)
WRITE (6,1870)
WRITE (6,1880)
IF (TMA.TL.EQ.2.) GO TO 1010
WRITE (6,1890)

```

QE,QEC  
FLOHP,FLOHPC  
THIE,THIEC  
THOE,THOEC  
FLONH3,FLNH3C  
PEVAP,PEVAPC  
T3A,T3AC  
T3,T3C  
X3P  
DSEVAP,DSEVPC  
TDOE,TDOECC  
TTE,TTEC  
TLER,TLERC  
IF (TMA.TL.EQ.2.) GO TO 1010



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A2205  
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A2207  
A2208

1010	GO TO 1020		
	CONTINUE		
	WRITE (6,1900)		
1020	CONTINUE		
	IF (PROF.EQ.2.) GO TO 1030		
	WRITE (6,1910)		
	GO TO 1040		
1030	CONTINUE		
	WRITE (6,1920)		
1040	CONTINUE		
	WRITE (6,1930) EPR		
	IF (EHFE.EQ.2.) GO TO 1050		
	WRITE (6,1940)		
	GO TO 1060		
1050	CONTINUE		
	WRITE (6,1950)		
1060	CONTINUE		
	WRITE (6,1960)	VSWE, VSWEC	
	WRITE (6,1970)	ETW2, ETW2C	
	WRITE (6,1980)	EFT, EFTC	
	WRITE (6,1990)	DELTAEC, DELTEC	
	WRITE (6,2000)	ELMTD, ELMTDC	
	WRITE (6,2010)	ELPSE	
	WRITE (6,2020)	ENTU	
	WRITE (6,2030)	UE, UEC	
	WRITE (6,2040)	HSWE, HSWEC	
	WRITE (6,2050)	HFSWE, HFSWEC	
	WRITE (6,2060)	HWE, HWEC	
	WRITE (6,2070)	HMH3E, HMH3EC	
	WRITE (6,2080)	THTAEC, THTAEC	
	WRITE (6,2090)	TSDE, TSDEC	
	WRITE (6,2100)	TNET	
	WRITE (6,2110)	DEVAP, DEVAPC	
	WRITE (6,2120)	P3, P3C	
	WRITE (6,2130)	T4, T4C	
	WRITE (6,2140)	X4P	
	WRITE (6,2150)	DSDEM, DSDEMC	
	IF (TYPEC.EQ.2.) GO TO 1070		
	WRITE (6,2160)		
	WRITE (6,2170)		
	GO TO 1080		
1070	CONTINUE		
	WRITE (6,2180)		
1080	CONTINUE		
	WRITE (6,2190)	QC, QCC	
	WRITE (6,2200)	FLQCP, FLOCP	
	WRITE (6,2210)	ITC, ICIC	
	WRITE (6,2220)	TCOC, TCOC	

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1090	WRITE (6,2230)	FLONH3,FLNH3C
	WRITE (6,2240)	PCOND,PCONDC
	WRITE (6,2250)	T5,T5C
	WRITE (6,2260)	T1,T1C
	WRITE (6,2270)	DSCOND,DSCONC
	WRITE (6,2280)	TDOC,TDOCCC
	WRITE (6,2290)	TTC,TTCC
	WRITE (6,2300)	TLCR,TLCRC
	WRITE (6,2310)	TLCR,TLCRC
	IF (TMA TL.EQ.2.) GO TO 1090	
	WRITE (6,2320)	
	GO TO 1100	
	CONTINUE	
1100	WRITE (6,2330)	
	CONTINUE	
	IF (PROF.EQ.2.) GO TO 1110	
	WRITE (6,2340)	
	GO TO 1120	
1110	CONTINUE	
	WRITE (6,2350)	
	CONTINUE	
1120	CONTINUE	
	IF (EHFC.EQ.2.) GO TO 1130	
	WRITE (6,2360)	CPR
	WRITE (6,2370)	
	GO TO 1140	
1130	CONTINUE	
	WRITE (6,2380)	
	CONTINUE	
1140	WRITE (6,2390)	VSWC,VSWCC
	WRITE (6,2400)	CTW2,CTW2C
	WRITE (6,2410)	CFI,CFIC
	WRITE (6,2420)	DELTA,DELTA
	WRITE (6,2430)	CLMTD,CLMTDC
	WRITE (6,2440)	EPSC
	WRITE (6,2450)	UC,UCC
	WRITE (6,2460)	UC,UCC
	WRITE (6,2470)	HSWC,HSMCC
	WRITE (6,2480)	HFSWC,HFSWCC
	WRITE (6,2490)	HWC,HWCC
	WRITE (6,2500)	HNH3C,HNH3CC
	WRITE (6,2510)	THTAC,THTACC
	WRITE (6,2520)	TSDC,TSDCC
	WRITE (6,2530)	TNCT
	WRITE (6,2540)	DCOND,DCONDC
	WRITE (6,2550)	
	WRITE (6,2560)	DIHP,DIHPC
	WRITE (6,2570)	TLHP,TLHPC
	WRITE (6,2580)	VSWHP,VSWHPC

WRITE	(6,2590)	FLOHP, FLOHPC	A2257
WRITE	(6,2600)	THIE, THIEC	A2258
WRITE	(6,2610)	CONH	A2259
WRITE	(6,2620)	DIPIH, DIPIHC	A2260
WRITE	(6,2630)	DICP, DICPC	A2261
WRITE	(6,2640)	TLCP, TLCPC	A2262
WRITE	(6,2650)	VSWCP, VSWCPC	A2263
WRITE	(6,2660)	FLOCP, FLOCPC	A2264
WRITE	(6,2670)	TCIC, TCICC	A2265
WRITE	(6,2680)	CONC	A2266
WRITE	(6,2690)	DPIPC, DPIPC	A2267
WRITE	(6,2700)	DINH3, DINH3C	A2268
WRITE	(6,2710)	TLNH3P, TLNH3PC	A2269
WRITE	(6,2720)	FLNH3, FLNH3C	A2270
WRITE	(6,2730)	DPIPN, DPIPNC	A2271
WRITE	(6,2740)	DINH3R, DINHRC	A2272
WRITE	(6,2750)	TLNHRP, TLNHRPC	A2273
WRITE	(6,2760)	DPIPNR, DPIPNRC	A2274
WRITE	(6,2770)	FLNHR, FLNHRPC	A2275
WRITE	(6,2780)	FLNHR, FLNHRPC	A2276
WRITE	(6,2790)	FLNHR, FLNHRPC	A2277
WRITE	(6,2800)	DPIPNR, DPIPNRC	A2278
WRITE	(6,2810)	DPMEC, DPMEC	A2279
WRITE	(6,2820)	QEPMP, QEPMPCC	A2280
WRITE	(6,2830)	EPE, EME	A2281
WRITE	(6,2840)	QEPMP, QEPMPCC	A2282
WRITE	(6,2850)	EPE, EME	A2283
WRITE	(6,2860)	DPMPCC, DPMPCC	A2284
WRITE	(6,2870)	QCPMP, QCPMPCC	A2285
WRITE	(6,2880)	EPC, EMC	A2286
WRITE	(6,2890)	DPMPCC, DPMPCC	A2287
WRITE	(6,2900)	QCPMP, QCPMPCC	A2288
WRITE	(6,2910)	DPMPNC, DPMPNC	A2289
WRITE	(6,2920)	QNPMP, QNPMPCC	A2290
WRITE	(6,2930)	EPNH3, EPNH3	A2291
WRITE	(6,2940)	DPMPNC, DPMPNC	A2292
WRITE	(6,2950)	QRPMP, QRPMPCC	A2293
WRITE	(6,2960)	EPNHR, EPNHR	A2294
WRITE	(6,2970)	EPNHR, EPNHR	A2295
WRITE	(6,2980)	EEP	A2296
WRITE	(6,2990)	ETRP	A2297
WRITE	(6,3000)	ETURBP	A2298
WRITE	(6,3010)	X5P	A2299
WRITE	(6,3020)	PWRTR, WELECG	A2300
WRITE	(6,3030)	WELOSS	A2301
WRITE	(6,3040)	PWREP, PWREPM	A2302
WRITE	(6,3050)		A2303
WRITE	(6,3060)		A2304

1150	WRITE	(6,3070)	PWRCP, PWRCPM	A2305
1160	WRITE	(6,3080)	PWRNP, PWRNPM	A2306
1170	WRITE	(6,3090)	PWRRP, PWRRM	A2307
1180	WRITE	(6,3100)	ELECT	A2308
1190	WRITE	(6,3110)	PPP	A2309
1200	WRITE	(6,3120)	TCE	A2310
1210	WRITE	(6,3130)		A2311
	WRITE	(6,3140)	CEVAP	A2312
	WRITE	(6,3150)	CCOND	A2313
	WRITE	(6,3160)	CTURB	A2314
	WRITE	(6,3170)	CGEN	A2315
	WRITE	(6,3180)	CCSWP	A2316
	WRITE	(6,3190)	CNH3P	A2317
	WRITE	(6,3200)	CNH3RP	A2318
	WRITE	(6,3210)		A2319
	WRITE	(6,3220)		A2320
	WRITE	(6,3230)	OBJ	A2321
	WRITE	(6,3240)	CPKW	A2322
	WRITE	(6,3250)		A2323
	RETURN			A2324
1220	FORMAT	(1H,19X,21H) INITIAL DESIGN VALUES		A2325
1230	FORMAT	(1H,2X,21H) HEVAPORATOR - HORIZONTAL		A2326
1240	FORMAT	(1H,2X,21H) HEVAPORATOR - VERTICAL		A2327
1250	FORMAT	(1H,4X,9H) TUBE O.D., 7X, F8.3, 4H(IN), 12X, F8.3, 4H(MM)		A2328
1260	FORMAT	(1H,4X,11H) TUBE LENGTH, 5X, F8.3, 4H(FT/S), 10X, F8.3, 5H(M/S)		A2329
1270	FORMAT	(1H,4X,11H) TUBE PRESSURE, 3X, F8.3, 9H(LBF/IN2), 7X, F8.3, 5H(MPA)		A2330
1280	FORMAT	(1H,4X,24H) TUBE MATERIAL - ALUMINUM, //, 6X, 15H THERMAL COND(K)		A2331
1290	FORMAT	(1H,4X,24H) TUBE MATERIAL - TITANIUM, //, 6X, 15H THERMAL COND(K)		A2332
1300	FORMAT	(1H,4X,37H) TUBE PROFILE - STAGGERED EQUI-LATERAL		A2333
1310	FORMAT	(1H,4X,33H) TUBE PROFILE - IN-LINE EQUI-SIDED		A2334
1320	FORMAT	(1H,6X,11H) PITCH RATIO, 6X, F5.2		A2335
1330	FORMAT	(1H,4X,24H) ENHANCEMENT - PLAIN TUBE		A2336
1340	FORMAT	(1H,4X,24H) ENHANCEMENT - LINDE-PROMOTER		A2337
1350	FORMAT	(1H,2X,20H) CONDENSER - HORIZONTAL		A2338
1360	FORMAT	(1H,2X,20H) CONDENSER - VERTICAL		A2339
	FORMAT	(1H,4X,9H) TUBE O.D., 7X, F8.3, 4H(IN), 12X, F8.3, 4H(MM)		A2340
	FORMAT	(1H,4X,11H) TUBE LENGTH, 5X, F8.3, 4H(FT/S), 10X, F8.3, 5H(M/S)		A2341
	FORMAT	(1H,4X,11H) TUBE PRESSURE, 3X, F8.3, 9H(LBF/IN2), 7X, F8.3, 5H(MPA)		A2342
	FORMAT	(1H,4X,24H) TUBE MATERIAL - ALUMINUM, //, 6X, 15H THERMAL COND(K)		A2343
	FORMAT	(1H,4X,24H) TUBE MATERIAL - TITANIUM, //, 6X, 15H THERMAL COND(K)		A2344

1370	11, F8.3, 13H(BTU/HR.FT.F), 3X, F8.3, 7H(W/M.CI)	A2353
1380	FORMAT (1H0.4X, 33HTUBE PROFILE - STAGGERED EQUI-LATERAL)	A2354
1390	FORMAT (1H0.4X, 33HTUBE PROFILE - IN-LINE REQ EQUI-SIDED)	A2355
1400	FORMAT (1H0.6X, 11HPITCH RATIO, 6X, F5.2)	A2356
1410	FORMAT (1H0.4X, 24HENHANCEMENT - PLAIN TUBE)	A2357
1420	FORMAT (1H0.4X, 28HENHANCEMENT - LINDE-PROMOTER)	A2358
1430	FORMAT (1H0.2X, 19HSALT WATER HOT PIPE)	A2359
1440	FORMAT (1H0.4X, 9HPIPE I.D., 7X, F8.3, 4H(FT), 12X, F8.3, 3H(M))	A2360
1450	FORMAT (1H0.4X, 11HPIPE LENGTH, 5X, F8.3, 4H(FT), 12X, F8.3, 3H(M))	A2361
1460	FORMAT (1H0.4X, 11HSW PIPE VEL, 5X, F8.3, 6H(FT/S), 10X, F8.3, 5H(M/S))	A2362
	FORMAT (1H0.4X, 13HSW INLET TEMP, 3X, F8.3, 7H(DEG F), 9X, F8.3, 7H(DEG C)	A2363
	FORMAT (1H0.4X, 12HSW SALINITY, 7X, F5.0, 1X, 5H0/000)	A2364
1470	FORMAT (1H0.2X, 20HSALT WATER COLD PIPE)	A2365
1480	FORMAT (1H0.4X, 9HPIPE I.D., 7X, F8.3, 4H(FT), 12X, F8.3, 3H(M))	A2366
1490	FORMAT (1H0.4X, 11HPIPE LENGTH, 5X, F8.3, 4H(FT), 12X, F8.3, 3H(M))	A2367
1500	FORMAT (1H0.4X, 11HSW PIPE VEL, 5X, F8.3, 6H(FT/S), 10X, F8.3, 5H(M/S))	A2368
1510	FORMAT (1H0.4X, 13HSW INLET TEMP, 3X, F8.3, 7H(DEG F), 9X, F8.3, 7H(DEG C)	A2369
1520	FORMAT (1H0.4X, 12HSW SALINITY, 7X, F5.0, 1X, 5H0/000)	A2370
1530	FORMAT (1H0.2X, 17HAMMONIA CIRC PIPE)	A2371
1540	FORMAT (1H0.4X, 9HPIPE I.D., 7X, F8.3, 4H(FT), 12X, F8.3, 3H(M))	A2372
1550	FORMAT (1H0.4X, 11HPIPE LENGTH, 5X, F8.3, 4H(FT), 12X, F8.3, 3H(M))	A2373
1560	FORMAT (1H0.2X, 20HAMMONIA RE-FLUX PIPE)	A2374
1570	FORMAT (1H0.4X, 9HPIPE I.D., 7X, F8.3, 4H(FT), 12X, F8.3, 3H(M))	A2375
1580	FORMAT (1H0.4X, 11HPIPE LENGTH, 5X, F8.3, 4H(FT), 12X, F8.3, 3H(M))	A2376
1590	FORMAT (1H0.2X, 29HPUMP AND GEN-TURB PERFORMANCE)	A2377
1600	FORMAT (1H0.4X, 12HEVAP SW PUMP)	A2378
1610	FORMAT (1H0.6X, 10HEFFICIENCY, 1X, 4HMECH, 2X, F5.2, 5H(PCT), 7X, 5HMOTOR,	A2379
1620	11X, F5.2, 5H(PCT))	A2380
1630	FORMAT (1H0.4X, 12HCOND SW PUMP)	A2381
1640	FORMAT (1H0.6X, 10HEFFICIENCY, 1X, 4HMECH, 2X, F5.2, 5H(PCT), 7X, 5HMOTOR,	A2382
	11X, F5.2, 5H(PCT))	A2383
1650	FORMAT (1H0.4X, 17HAMMONIA CIRC PUMP)	A2384
1660	FORMAT (1H0.6X, 10HEFFICIENCY, 1X, 4HMECH, 2X, F5.2, 5H(PCT), 7X, 5HMOTOR,	A2385
	11X, F5.2, 5H(PCT))	A2386
1670	FORMAT (1H0.2X, 21HGEN-TURB EFFICIENCIES)	A2387
1680	FORMAT (1H0.6X, 14HGEN MECH/ELECT, 3X, F5.2, 5H(PCT))	A2388
1690	FORMAT (1H0.6X, 9HTURB MECH, 8X, F5.2, 5H(PCT))	A2389
1700	FORMAT (1H0.2X, 18HPNET POWER REQUIREMENTS)	A2390
1710	FORMAT (1H1.19HNET POWER OUTPUT, 10X, F8.3, 4H(MW))	A2391
1720	FORMAT (1H0.2X, 23HEVAPORATOR - HORIZONTAL)	A2392
1730	FORMAT (1H0.4X, 9HHT ABSORB, 1X, F14.1, 8H(BTU/HR), 7X, F8.3, 4H(MW))	A2393
1740	FORMAT (1H0.4X, 7HSW FLOW, 3X, F14.1, 8H(LBM/HR), 1X, F14.1, 7H(KG/HR))	A2394
1750	FORMAT (1H0.6X, 10HSW TEMP IN, 4X, F8.3, 7H(DEG F), 8X, F8.3, 7H(DEG C))	A2395
1760	FORMAT (1H0.6X, 11HSW TEMP OUT, 3X, F8.3, 7H(DEG F), 8X, F8.3, 7H(DEG C))	A2396
1770	FORMAT (1H0.6X, 11HSW TEMP OUT, 3X, F8.3, 7H(DEG F), 8X, F8.3, 7H(DEG C))	A2397
1780	FORMAT (1H0.6X, 11HSW TEMP OUT, 3X, F8.3, 7H(DEG F), 8X, F8.3, 7H(DEG C))	A2398
	FORMAT (1H0.6X, 11HSW TEMP OUT, 3X, F8.3, 7H(DEG F), 8X, F8.3, 7H(DEG C))	A2399
	FORMAT (1H0.6X, 11HSW TEMP OUT, 3X, F8.3, 7H(DEG F), 8X, F8.3, 7H(DEG C))	A2400

1790	FORMAT	(1H0,4X,8HMH3 FLOW,2X,F14.1,8H(LBM/HR),1X,F14.1,7H(KG/HR))	A2401
1800	FORMAT	(1H0,4X,13HOPR PRESSURE,3X,F8.3,9H(LBF/IN2),6X,F8.3,5H(KPA	A2402
	1))		A2403
1810	FORMAT	(1H0,6X,13HEVAP SAT TEMP,1X,F8.3,7H(DEG F),8X,F8.3,7H(DEG C	A2404
	1))		A2405
1820	FORMAT	(1H0,6X,11HOUTLET TEMP,3X,F8.3,7H(DEG F),8X,F8.3,7H(DEG C))	A2406
1830	FORMAT	(1H0,6X,14HOUTLET QUALITY,4X,F5.2,5H(PCT))	A2407
1840	FORMAT	(1H0,4X,14HMH3 PRESS DROP,2X,F8.3,9H(LBF/IN2),6X,F8.3,5H(KP	A2408
	1A))		A2409
1850	FORMAT	(1H0,4X,20HTUBE CHARACTERISTICS)	A2410
1860	FORMAT	(1H0,6X,10HOUTTER DIA,4X,F8.3,4H(IN),11X,F8.3,4H(MM))	A2411
1870	FORMAT	(1H0,6X,10HWALL THICK,4X,F8.3,4H(IN),11X,F8.3,4H(MM))	A2412
1880	FORMAT	(1H0,6X,6HLENGTH,8X,F8.3,4H(F),11X,F8.3,3H(M))	A2413
1890	FORMAT	(1H0,6X,19HMATERIAL - ALUMINUM)	A2414
1900	FORMAT	(1H0,6X,19HMATERIAL - TITANIUM)	A2415
1910	FORMAT	(1H0,6X,37HTUBE PROFILE - STAGGERED EQUI-LATERAL)	A2416
1920	FORMAT	(1H0,6X,37HTUBE PROFILE - IN-LINE EQUI-SIDED)	A2417
1930	FORMAT	(1H0,8X,11HPITCH RATIO,4X,F5.2)	A2418
1940	FORMAT	(1H0,6X,24HENHANCEMENT - PLAIN TUBE)	A2419
1950	FORMAT	(1H0,6X,28HENHANCEMENT - LINDE-PROMOTER)	A2420
1960	FORMAT	(1H0,6X,11HSW VELOCITY,3X,F8.3,6H(FT/S),9X,F8.3,5H(M/S))	A2421
1970	FORMAT	(1H0,4X,17HT WALL(SHELLSIDE),F7.3,7H(DEG F),8X,F8.3,7H(DEG	A2422
	1C))		A2423
1980	FORMAT	(1H0,4X,9HFILM TEMP,7X,F8.3,7H(DEG F),8X,F8.3,7H(DEG C))	A2424
1990	FORMAT	(1H0,4X,15HDELTA T BOILING,1X,F8.3,7H(DEG F),8X,F8.3,7H(DEG	A2425
	1C))		A2426
2000	FORMAT	(1H0,4X,8HL.M.T.D.8X,F8.3,7H(DEG F),8X,F8.3,7H(DEG C))	A2427
2010	FORMAT	(1H0,4X,18HEVAP EFFECTIVENESS,7X,F8.3)	A2428
2020	FORMAT	(1H0,4X,20HNR OF TRANSFER UNITS,5X,F8.3)	A2429
2030	FORMAT	(1H0,4X,11HCVL HT COEF,5X,F8.2,14H(BTU/HR.FT2.F),1X,F8.2,8H	A2430
	1(W/M2.C))		A2431
2040	FORMAT	(1H0,6X,8HH(WATER),6X,F8.2,14H(BTU/HR.FT2.F),1X,F8.2,8H(W/M	A2432
	12.C))		A2433
2050	FORMAT	(1H0,6X,10HH(FOULING),4X,F8.2,14H(BTU/HR.FT2.F),1X,F8.2,8H(	A2434
	1W/M2.C))		A2435
2060	FORMAT	(1H0,6X,8HH(METAL),6X,F8.2,14H(BTU/HR.FT2.F),F9.2,8H(W/M2.C	A2436
	1))		A2437
2070	FORMAT	(1H0,6X,10HH(AMMONIA),4X,F8.2,14H(BTU/HR.FT2.F),1X,F8.2,8H(	A2438
	1W/M2.C))		A2439
2080	FORMAT	(1H0,4X,10HHT SURFACE,2X,F12.2,5H(FT2),6X,F12.2,4H(M2))	A2440
2090	FORMAT	(1H0,4X,14HTUBE SHEET DIA,2X,F8.3,4H(F),11X,F8.3,3H(M))	A2441
2100	FORMAT	(1H0,6X,15HTOT NR OF TUBES,7X,F12.0)	A2442
2110	FORMAT	(1H0,4X,13HSW PRESS DROP,3X,F8.3,9H(LBF/IN2),6X,F8.3,5H(KPA	A2443
	1))		A2444
2120	FORMAT	(1H0,2X,36HMOISTURE SEPARATOR-INSIDE EVAP SHELL)	A2445
2130	FORMAT	(1H0,4X,13HOPR PRESSURE,3X,F8.3,9H(LBF/IN2),6X,F8.3,5H(KPA	A2446
	1))		A2447
2140	FORMAT	(1H0,6X,11HOUTLET TEMP,3X,F8.3,7H(DEG F),8X,F8.3,7H(DEG C))	A2448

2150	FORMAT	(1H0,6X,14HOUTLET QUALITY,3X,F5.2,5H(PCT))	A2449
2160	FORMAT	(1H0,6X,14HNNH3 PRESS DROP,F8.3,9H(LBF/IN2),6X,F8.3,5H(KPA))	A2450
2170	FORMAT	(1H1,2X,22HCONDENSER - HORIZONTAL)	A2451
2180	FORMAT	(1H1,2X,22HCONDENSER - VERTICAL)	A2452
2190	FORMAT	(1H0,4X,9HHT REJECT,1X,F14.1,8H(BTU/HR),7X,F8.3,4H(MW))	A2453
2200	FORMAT	(1H0,4X,7HSM FLOW,3X,F14.1,8H(LBM/HR),1X,F14.1,7H(KG/HR))	A2454
2210	FORMAT	(1H0,6X,10HSM TEMP IN,4X,F8.3,7H(DEG F),8X,F8.3,7H(DEG C))	A2455
2220	FORMAT	(1H0,6X,11HSM TEMP OUT,3X,F8.3,7H(DEG F),8X,F8.3,7H(DEG C))	A2456
2230	FORMAT	(1H0,4X,8HNNH3 FLOW,2X,F14.1,8H(LBM/HR),1X,F14.1,7H(KG/HR))	A2457
2240	FORMAT	(1H0,4X,13HOPER PRESSURE,3X,F8.3,9H(LBF/IN2),6X,F8.3,5H(KPA))	A2458
	1)		A2459
2250	FORMAT	(1H0,6X,13HCOND SAT TEMP,1X,F8.3,7H(DEG F),8X,F8.3,7H(DEG C))	A2460
	1)		A2461
2260	FORMAT	(1H0,6X,11HOUTLET TEMP,3X,F8.3,7H(DEG F),8X,F8.3,7H(DEG C))	A2462
2270	FORMAT	(1H0,4X,14HNNH3 PRESS DROP,2X,F8.3,9H(LBF/IN2),6X,F8.3,5H(KPA))	A2463
	1A)		A2464
2280	FORMAT	(1H0,4X,20HTUBE CHARACTERISTICS)	A2465
2290	FORMAT	(1H0,6X,10HOUTTER DIA,4X,F8.3,4H(IN),11X,F8.3,4H(MM))	A2466
2300	FORMAT	(1H0,6X,10HWALL THICK,4X,F8.3,4H(IN),11X,F8.3,4H(MM))	A2467
2310	FORMAT	(1H0,6X,6HLENGTH,8X,F8.3,4H(FT),11X,F8.3,3H(M))	A2468
2320	FORMAT	(1H0,6X,19HMATERIAL - ALUMINUM)	A2469
2330	FORMAT	(1H0,6X,19HMATERIAL - TITANIUM)	A2470
2340	FORMAT	(1H0,6X,33HTUBE PROFILE - STAGGERED EQUI-LATERAL)	A2471
2350	FORMAT	(1H0,6X,33HTUBE PROFILE - IN-LINE EQUI-SIDED)	A2472
2360	FORMAT	(1H0,8X,11HPITCH RATIO,4X,F5.2)	A2473
2370	FORMAT	(1H0,6X,24HENHANCEMENT - PLAIN-TUBE)	A2474
2380	FORMAT	(1H0,6X,28HENHANCEMENT - LINDE-PROMOTER)	A2475
2390	FORMAT	(1H0,6X,11HSM VELOCITY,3X,F8.3,6H(FT/S),9X,F8.3,5H(M/S))	A2476
2400	FORMAT	(1H0,4X,17HT WALL(SHELLSIDE),F7.3,7H(DEG F),8X,F8.3,7H(DEG C))	A2477
	1C)		A2478
2410	FORMAT	(1H0,4X,9HFILM TEMP,7X,F8.3,7H(DEG F),8X,F8.3,7H(DEG C))	A2479
2420	FORMAT	(1H0,4X,12HDELTA T COND,4X,F8.3,7H(DEG F),8X,F8.3,7H(DEG C))	A2480
	1)		A2481
2430	FORMAT	(1H0,4X,8HL-M.T.D.,8X,F8.3,7H(DEG F),8X,F8.3,7H(DEG C))	A2482
2440	FORMAT	(1H0,4X,18HCOND EFFECTIVENESS,7X,F8.3)	A2483
2450	FORMAT	(1H0,4X,20HNR OF TRANSFER UNITS,5X,F8.3)	A2484
2460	FORMAT	(1H0,4X,11HOVL HT COEF,5X,F8.2,14H(BTU/HR.FT2.F),1X,F8.2,8H(W/M2.C))	A2485
	1W/M2.C)		A2486
2470	FORMAT	(1H0,6X,8HH(WATER),6X,F8.2,14H(BTU/HR.FT2.F),1X,F8.2,8H(W/M2.C))	A2487
	12.C)		A2488
2480	FORMAT	(1H0,6X,10HH(FOULING),4X,F8.2,14H(BTU/HR.FT2.F),1X,F8.2,8H(W/M2.C))	A2489
2490	FORMAT	(1H0,6X,8HH(METAL),6X,F8.2,14H(BTU/HR.FT2.F),F9.2,8H(W/M2.C))	A2490
	1)		A2491
2500	FORMAT	(1H0,6X,10HH(AMMONIA),4X,F8.2,14H(BTU/HR.FT2.F),1X,F8.2,8H(W/M2.C))	A2492
	1W/M2.C)		A2493
2510	FORMAT	(1H0,4X,10HHT SURFACE,2X,F12.2,5H(FT2),6X,F12.2,4H(M2))	A2494
2520	FORMAT	(1H0,4X,14HTUBE SHEET DIA,2X,F8.3,4H(FT),11X,F8.3,3H(M))	A2495
			A2496

2530	FORMAT	{1H0,6X,15HSW NR OF TUBES,7X,F12.0}		A2497
2540	FORMAT	{1H0,4X,13HSW PRESS DROP,3X,F8.3,9H(LBF/IN2),6X,F8.3,5H(KPA		A2498
	1)}			A2499
2550	FORMAT	{1H1,2X,19HSALT WATER HOT PIPE}		A2500
2560	FORMAT	{1H0,4X,9HPPIPE I.D.,7X,F8.3,4H(FT),11X,F8.3,3H(M)}		A2501
2570	FORMAT	{1H0,4X,11HPPIPE LENGTH,5X,F8.3,4H(FT),11X,F8.3,3H(M)}		A2502
2580	FORMAT	{1H0,4X,11HSW PIPE VEL,5X,F8.3,6H(FT/S),9X,F8.3,5H(M/S)}		A2503
2590	FORMAT	{1H0,4X,7HSW FLOW,3X,F14.1,8H(LBM/HR),1X,F14.1,7H(KG/HR)}		A2504
2600	FORMAT	{1H0,4X,13HSW INLET TEMP,3X,F8.3,7H(DEG F),8X,F8.3,7H(DEG C		A2505
	1)}			A2506
2610	FORMAT	{1H0,4X,12HSW SALINITY,7X,F5.0,1X,5H0/000}		A2507
2620	FORMAT	{1H0,4X,13HSW PRESS DROP,3X,F8.3,9H(LBF/IN2),6X,F8.3,5H(KPA		A2508
	1)}			A2509
2630	FORMAT	{1H0,2X,20HSALT WATER COLD PIPE}		A2510
2640	FORMAT	{1H0,4X,9HPPIPE I.D.,7X,F8.3,4H(FT),11X,F8.3,3H(M)}		A2511
2650	FORMAT	{1H0,4X,11HPPIPE LENGTH,5X,F8.3,4H(FT),11X,F8.3,3H(M)}		A2512
2660	FORMAT	{1H0,4X,11HSW PIPE VEL,5X,F8.3,6H(FT/S),9X,F8.3,5H(M/S)}		A2513
2670	FORMAT	{1H0,4X,7HSW FLOW,3X,F14.1,8H(LBM/HR),1X,F14.1,7H(KG/HR)}		A2514
2680	FORMAT	{1H0,4X,13HSW INLET TEMP,3X,F8.3,7H(DEG F),8X,F8.3,7H(DEG C		A2515
	1)}			A2516
2690	FORMAT	{1H0,4X,12HSW SALINITY,7X,F5.0,1X,5H0/000}		A2517
2700	FORMAT	{1H0,4X,13HSW PRESS DROP,3X,F8.3,9H(LBF/IN2),6X,F8.3,5H(KPA		A2518
	1)}			A2519
2710	FORMAT	{1H0,2X,17HAMMONIA CIRC PIPE}		A2520
2720	FORMAT	{1H0,4X,9HPPIPE I.D.,7X,F8.3,4H(FT),11X,F8.3,3H(M)}		A2521
2730	FORMAT	{1H0,4X,11HPPIPE LENGTH,5X,F8.3,4H(FT),11X,F8.3,3H(M)}		A2522
2740	FORMAT	{1H0,4X,8HNNH3 FLOW,2X,F14.1,8H(LBM/HR),1X,F14.1,7H(KG/HR)}		A2523
2750	FORMAT	{1H0,4X,14HNNH3 PRESS DROP,2X,F8.3,9H(LBF/IN2),6X,F8.3,5H(KP		A2524
	1A)}			A2525
2760	FORMAT	{1H0,2X,25HAMMONIA RE-FLUX CIRC PIPE}		A2526
2770	FORMAT	{1H0,4X,9HPPIPE I.D.,7X,F8.3,4H(FT),11X,F8.3,3H(M)}		A2527
2780	FORMAT	{1H0,4X,11HPPIPE LENGTH,5X,F8.3,4H(FT),11X,F8.3,3H(M)}		A2528
2790	FORMAT	{1H0,4X,8HNNH3 FLOW,2X,F14.1,8H(LBM/HR),1X,F14.1,7H(KG/HR)}		A2529
2800	FORMAT	{1H0,4X,14HNNH3 PRESS DROP,2X,F8.3,9H(LBF/IN2),6X,F8.3,5H(KP		A2530
	1A)}			A2531
2810	FORMAT	{1H1,2X,29HPUMP AND GEN-TURB PERFORMANCE}		A2532
2820	FORMAT	{1H0,4X,12HEVAP SW PUMP}		A2533
2830	FORMAT	{1H0,6X,10HHEAD PRESS,4X,F8.3,4H(FT),11X,F8.3,3H(M)}		A2534
2840	FORMAT	{1H0,6X,8HCAPACITY,2X,F12.1,9H(GAL/MIN),2X,F12.1,9H(LIT/MIN		A2535
	1)}			A2536
2850	FORMAT	{1H0,6X,10HEFFICIENCY,1X,4HMECH,2X,F5.2,5H(PCT),7X,5HMOTOR,		A2537
	1X,F5.2,5H(PCT)}			A2538
2860	FORMAT	{1H0,4X,12HCOND SW PUMP}		A2539
2870	FORMAT	{1H0,6X,10HCOND PRESS,4X,F8.3,4H(FT),11X,F8.3,3H(M)}		A2540
2880	FORMAT	{1H0,6X,8HCAPACITY,2X,F12.1,9H(GAL/MIN),2X,F12.1,9H(LIT/MIN		A2541
	1)}			A2542
2890	FORMAT	{1H0,6X,10HEFFICIENCY,1X,4HMECH,2X,F5.2,5H(PCT),7X,5HMOTOR,		A2543
	1X,F5.2,5H(PCT)}			A2544



2900	FORMAT	(1H0,4X,17HAMMONIA CIRC PUMP)	A2545
2910	FORMAT	(1H0,6X,10HHEAD PRESS,4X,F8.3,4H(FT),11X,F8.3,3H(M))	A2546
2920	FORMAT	(1H0,6X,8HCAPACITY,2X,F12.1,9H(GAL/MIN),2X,F12.1,9H(LIT/MIN	A2547
	1))		A2548
2930	FORMAT	(1H0,6X,10HEFFICIENCY,1X,4HMECH,2X,F5.2,5H(PCT),7X,5HMOTOR,	A2549
	11X,F5.2,5H(PCT))		A2550
2940	FORMAT	(1H0,4X,20HAMMONIA RE-FLUX PUMP)	A2551
2950	FORMAT	(1H0,6X,10HHEAD PRESS,4X,F8.3,4H(FT),11X,F8.3,3H(M))	A2552
2960	FORMAT	(1H0,6X,8HCAPACITY,2X,F12.1,9H(GAL/MIN),2X,F12.1,9H(LIT/MIN	A2553
	1))		A2554
2970	FORMAT	(1H0,6X,10HEFFICIENCY,1X,4HMECH,2X,F5.2,5H(PCT),7X,5HMOTOR,	A2555
	11X,F5.2,5H(PCT))		A2556
2980	FORMAT	(1H0,4X,21HGEN-TURB EFFICIENCIES)	A2557
2990	FORMAT	(1H0,6X,14HGEN MECH&ELECT,4X,F5.2,5H(PCT))	A2558
3000	FORMAT	(1H0,6X,9HTURB MECH,9X,F5.2,5H(PCT))	A2559
3010	FORMAT	(1H0,6X,13HTURB INTERNAL,5X,F5.2,5H(PCT))	A2560
3020	FORMAT	(1H0,4X,19HTURB OUTLET QUALITY,1X,F5.2,5H(PCT))	A2561
3030	FORMAT	(1H1,2X,18HPOWER REQUIREMENTS)	A2562
3040	FORMAT	(1H0,4X,14HTURB-GEN GROSS,F10.3,4H(HPI),12X,F8.3,4H(MW))	A2563
3050	FORMAT	(1H0,6X,17HEFFICIENCY LOSSES,21X,F8.3,4H(MW))	A2564
3060	FORMAT	(1H0,4X,12HEVAP SW PUMP,4X,F8.3,4H(HPI),12X,F8.3,4H(MW))	A2565
3070	FORMAT	(1H0,4X,12HCOND SW PUMP,4X,F8.3,4H(HPI),12X,F8.3,4H(MW))	A2566
3080	FORMAT	(1H0,4X,13HNNH3 CIRC PUMP,3X,F8.3,4H(HPI),12X,F8.3,4H(MW))	A2567
3090	FORMAT	(1H0,4X,16HNNH3 RE-FLUX PUMP,F8.3,4H(HPI),12X,F8.3,4H(MW))	A2568
3100	FORMAT	(1H0,42X,10H	A2569
		NET POWER-OUTPUT,18X,F8.3,4H(MW))	A2570
3110	FORMAT	(1H0,2X,23HPERCENT PARASITIC POWER,10X,F5.2,5H(PCT))	A2571
3120	FORMAT	(1H0,2X,29HTHEODYNAMIC CYCLE EFFICIENCY,4X,F5.2,5H(PCT))	A2572
3130	FORMAT	(1H0,2X,18HCOST OF COMPONENTS)	A2573
3140	FORMAT	(1H0,4X,10HEVAPORATOR,14X,F12.2,9H(DOLLARS))	A2574
3150	FORMAT	(1H0,4X,9HCONDENSER,12X,F12.2,9H(DOLLARS))	A2575
3160	FORMAT	(1H0,4X,11HGEN-TURBINE,13X,F12.2,9H(DOLLARS))	A2576
3170	FORMAT	(1H0,4X,11HGENERATOR,15X,F12.2,9H(DOLLARS))	A2577
3180	FORMAT	(1H0,4X,12HEVAP SW PUMP,12X,F12.2,9H(DOLLARS))	A2578
3190	FORMAT	(1H0,4X,12HCOND SW PUMP,12X,F12.2,9H(DOLLARS))	A2579
3200	FORMAT	(1H0,4X,13HNNH3 CIRC PUMP,11X,F12.2,9H(DOLLARS))	A2580
3210	FORMAT	(1H0,4X,16HNNH3 RE-FLUX PUMP,8X,F12.2,9H(DOLLARS))	A2581
3220	FORMAT	(1H0,28X,12H	A2582
3230	FORMAT	(1H0,10X,12HOPTIMUM COST,6X,F12.2,9H(DOLLARS))	A2583
3240	FORMAT	(1H0,2X,22HCOST PER NET KW OUTPUT,4X,F12.2,17H(DOLLARS)	A2584
3250	FORMAT	1) END	A2585

C FUNCTION SUBROUTINES FOR SALT WATER AND AMMONIA  
C SALT WATER DENSITY(LBM/FT3)  
C  
FUNCTION RHOSW(IT)  
RHOSW=63.90152+.017512\*T-5.684E-04\*T\*\*2+.6.336E-06\*T\*\*3-2.7949E-08\*  
IT\*\*4

[illegible]

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C      FUNCTION VSNH3 (I)
C      VSNH3=;55203968261-.003405489426*I+1.01587304E-05*I**2-1.5046298E-
C      108*I**3
C      RETURN
C      END
C      SPECIFIC HEAT OF SAT AMMONIA LIQUID(BTU/LBM.F)
C      FUNCTION CPNH3 (I)
C      CPNH3=1.0829682542+.00041058198*I+1.7658742E-06*I**2+1.6203699E-08
C      1*I**3
C      RETURN
C      END
C      THERMAL CONDUCTIVITY OF SAT AMMONIA LIQ(BTU/HR.FT.F)
C      FUNCTION TKNH3 (I)
C      TKNH3=.33517857143-.000720535715*I-2.2321429E-07*I**2
C      RETURN
C      END
C      ENTROPY OF SAT AMMONIA LIQUID(BTU/LBM.R)
C      FUNCTION SF (I)
C      SF=.09771914+.002334056*I-1.66882E-06*I**2+2.1649E-09*I**3
C      RETURN
C      END
C      ENTROPY OF SAT AMMONIA VAPOR(BTU/LBM.R)
C      FUNCTION SG (I)
C      SG=1.33473105-.001973293*I+4.0808E-06*I**2-7.5887E-09*I**3
C      RETURN
C      END
C      DENSITY OF SAT AMMONIA LIQ(LBM/FT3)
C      FUNCTION RFNH3 (I)
C      RFNH3=41.415499-.0490092*I+.000039001*I**2-5.2019E-07*I**3
C      RETURN
C      END
C      DENSITY OF SAT AMMONIA VAPOR(LBM/FT3)
C      FUNCTION RGNH3 (P)
C      RGNH3=.01242535+.0032785773*P-2.52513E-07*P**2+8.9285E-10*P**3
C      RETURN
C      END
C      TITANIUM TUBE WALL THICKNESS - PLAIN(INCH)
C      FUNCTION TP (D)
C      TP=.007966159+.040237327*D-.035418004*D**2+.015039976*D**3-.002418
C      12547*D**4

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3 4 5 6-  
1 2 3 4 5 6-  
Y Y Y Y Z Z Z Z Z

FUNCTION AEC (D)  
AEC=2.157473-5.206014\*D+7.336678\*D\*\*2-3.656495\*D\*\*3+.7204888\*D\*\*4  
RETURN  
END  
COND ALUMINUM TUBE COST/FT - LINDE-PROMOTER(\$/FT)  
FUNCTION ACC (D)  
ACC=1.799865-4.543195\*D+6.56578\*D\*\*2-3.305876\*D\*\*3+.6622916\*D\*\*4  
RETURN  
END

C  
C

## LIST OF REFERENCES

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